Ford Edge Drivetrain Pulse Reduction

ME 450, Section 3, Winter 2008

Final Report

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EXECUTIVE SUMMARY

During a full-frontal impact into a rigid barrier at 35 mph, the all-wheel drive (AWD) Ford Edge and AWD Lincoln MKX experienced G-force deceleration pulses in excess of 60 G's. The average deceleration pulses for a front-wheel drive (FWD) Edge range from 30-40 G's, and the major difference between an AWD and FWD vehicle is the addition of the drivetrain, which consists of the power take-off (PTO), driveshaft, and rear differential unit (RDU). It was originally determined that the engine roll restrictor was not deforming during impact and thus acted as a direct load path, but further analysis led to the conclusion that the impact between the RDU mounting brackets and the rear subframe was the largest contributor to the pulse. The goal of this project was to reduce the drivetrain-related deceleration pulses in the vehicle by redesigning the rear RDU mounting brackets to lessen the severity of the impact between the RDU and rear subframe.

The redesigned RDU attachment brackets had to be compatible with the current RDU and subframe while preserving the functionality of the current brackets. Due to the late changes in the project, we were asked to provide alpha designs of the brackets in order to prove a concept instead of full prototypes, which were fabricated at the Livernois Vehicle Development Center. These brackets were installed on a Ford Edge which then underwent an NCAP test and was compared to CAE simulation models and an A-to-B comparison with the latest NCAP Lincoln MKX test.

The model predicted a decrease of 9-15 G's in the drivetrain-related pulse, and we greatly exceeded this expectation by recording a 26 G drop in the peak pulse due to the modifications in the brackets. These changes increased the clearances between the RDU and rear subframe and provided the driveshaft with the displacements needed to disengage, thus decreasing the load transfer to the RDU and subframe.

Due to the shortened timeline of this project, the brackets were designed for crash safety, which leaves open issues for NVH (Noise, Vibrations, and Harshness), vehicle dynamics, and durability. We recommend presenting our final design and findings to the RDU mounting bracket design and release (D&R) engineer so that the design can be refined to meet all functional vehicle attributes.

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INTRODUCTION

The National Highway Traffic Safety Association rigorously tests new vehicles to ensure their crashworthiness and assign star ratings. The New Car Assessment Program (NCAP) involves a full-frontal impact into a rigid barrier at 35 mph, which simulates a 70 mph head-on crash between two vehicles.

During a summer internship, one of the team members was asked to investigate the root cause of the high G-force deceleration pulses experienced by the Ford Edge and Lincoln MKX. The pulses are often caused by internal impacts within the vehicle as components buckle and crush, and these pulses are detected by accelerometers throughout the automotive structure. High deceleration pulses in the body frame lead to higher loads experienced by the passengers in the vehicle. The main focus in the Ford Edge investigation was concentrated on the pulse related to the drivetrain, which includes the power take-off unit, drive shaft, engine roll restrictor, and rear differential unit (RDU). This high pulse was measured at the rocker at the B-pillar, which is located below and behind the driver's side door.

It was initially determined that the engine roll restrictor was creating a direct load path as it bottomed out between the transmission, power take-off, and drive shaft. Furthermore, the roll restrictor does not deform during the impact, but remains stationary until the transmission case to which it is attached fractures. These observations were presented to Steve Kozak, the Chief Safety Engineer at Ford Motor Company, as well as the recommendation that the roll restrictor be redesigned to disrupt the direct transfer of the forces through the subframe of the vehicle. Mr. Kozak then requested that the work started over the summer be integrated into a senior design project for the University of Michigan.

The initial focus of the project was shifted to redesigning the RDU brackets mounted to the subframe due to a number of reasons. First, upon running a CAE model with and without the roll restrictor present, it was determined that the effects of the roll restrictor to the overall G-force deceleration pulse was not substantial enough to warrant the redesign. Second, the roll restrictor bracket has been redesigned for vehicle dynamics, as well as for cost savings, and in doing so, created a conflict for our goal of weakening the bracket to disrupt the load path. Finally, upon close examination of a recent NCAP post-crash vehicle, it was observed that decoupling or deformation of the roll restrictor would not sufficiently disrupt the load path since the engine and transmission case's movement is greatly limited by the undercarriage design in addition to the roll restrictor.

After further analysis of post-crash footage and a hoist review at the Ford Crash Barrier Building, it was determined that the interaction between the RDU and the rear subframe substantially contributed to high deceleration pulses, and the focus of our project should be redirected to redesigning the rear RDU mounting brackets. Figure 1, page 5, shows the RDU connected to the subframe with the brackets circled.

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Figure 1. CAE rendering of RDU on rear subframe.

The goal of this project was to redesign the RDU attachments to the subframe so that the severity of the impact between the two components is reduced and the deceleration pulse due to the effect of the drivetrain is reduced. Our team worked with engineers at Ford to redesign the RDU mounts and with technicians at Livernois to machine the new attachments. A proof of concept was tested on a full vehicle in an NCAP test at the Ford Barrier and an alpha design was determined based upon those results.

PROBLEM DESCRIPTION

It was initially determined that the engine roll restrictor was creating a direct load path as it bottomed out between the transmission, power take-off, and driveshaft, creating a large contribution to the deceleration pulse. However, upon running a CAE model with and without the roll restrictor present, it was determined that the contribution of the roll restrictor was not substantial enough to warrant a redesign. Also, the roll restrictor bracket has been redesigned for vehicle dynamics, as well as for cost savings, and in doing so, created a conflict for our goal of weakening the bracket as the redesign made the bracket sturdier. Finally, upon close examination of a recent NCAP post-crash vehicle, it was observed that decoupling or deformation of the roll restrictor would not sufficiently disrupt the load path since the engine and transmission case's movement is greatly limited by the undercarriage design in addition to the roll restrictor. Design Review #2, which reports all work done on the project relating to the roll restrictor, can be found in Appendix A. Appendices to Design Review #2 are not included.

To reduce the contribution of the RDU to the deceleration pulses, the interaction between the RDU and subframe must be decreased. This reduction was achieved by increasing the travel between the RDU subframe attachments and the rear subframe. The new attachments must be compatible with the current undercarriage design, including the rear subframe and RDU, which is used on multiple Ford vehicles.

The main focus of these new attachments was to preserve their function (support the RDU and retain the connection between the RDU and subframe) while decreasing the negative impact on crashworthiness.

Since the focus of our project was redirected halfway through the project schedule, our sponsor was more interested in a proof of concept than a production-ready part. Therefore, our design to prove the concept was fabricated onsite at the Livernois facility where the vehicle was prepped for an NCAP test.

Due to customer confidentiality, most of our work needed to be done in conjunction with Ford Motor Company engineers, including Computer Aided Engineering (CAE) at the Dearborn facilities. We formed contacts with engineers at Ford to be able to complete our analyses. Also, since we do not have the capabilities to do so, vehicle modifications was carried out by the Ford Motor Company's Livernois facility.

PRIOR WORK

During the summer of 2007, the effects of the drivetrain on the G-force deceleration pulse were investigated, and the contributions of the driveshaft and RDU were considered. Analyses of crash videos and data were conducted, as well as hoist reviews of post-crash vehicles, and from this, component-level tests were conducted to determine the individual contributions of each element.

The drivetrain on a Ford Edge AWD vehicle is the same as that of a FWD vehicle, with the addition of the power take-off (PTO), driveshaft, and RDU. The PTO is located behind the transmission above the front subframe crossmember and transfers rotational motion through the driveshaft to the RDU, which subsequently transfers the motion to the rear axle. These three components are shown below in the undercarriage of an Edge in Figure 2 below.

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Figure 2. Undercarriage of Edge with PTO, driveshaft, and RDU called out.

The original driveshaft (pre-2009 model year) consisted of a three-piece design, but the current design is two pieces and will be considered in the current analysis. The driveshaft was connected to the undercarriage in the front at the power take-off (PTO), in the rear at the RDU, and at two locations on the undercarriage using brackets that contained keyways. These keyways were designed to break away during impact due to the rearward motion of the driveshaft relative to the vehicle, allowing the driveshaft to drop vertically and disrupt the load path during the crash. Between each segment, a crash disk was placed to encourage the movement of the brackets to open the keyway as seen in Figure 3 through Figure 6, page 7-8. A component-level test was designed to determine the forces and displacements required to cause the keyways to open and the driveshaft to come free. These tests were conducted on a quasi-static testing apparatus (Tinius Olsen) and provided loads and displacements to aid in further analysis.

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Figure 3. Pre-test picture of driveshaft setup on Tinius Olsen facility

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Figure 4. Post-test picture of driveshaft with three keyways disengaged

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Figure 5. Original keyway design

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Figure 6. Alternate concept with larger keyways

From reviewing the crash test videos, significant movement of the RDU was noted. Due to limited camera views, hoist reviews were conducted to examine the post-crash vehicle underbody. The hoist review revealed fractures on the RDU casing and witness marks on the subframe, and the bushings in the rear subframe were pushed out, showing the severity of the impact between the components. Component-level testing of the RDU was completed to replicate the impact observed between the RDU and rear subframe. Contact switches were used to detect the timing of impact. The test setup is shown below in Figure 7.

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Figure 7. RDU component level testing

From video and dimensional analysis of CAE models, it was determined that the time and crush displacement needed to force the breakaway of the driveshaft keyways was not enough to prevent the impact of the RDU to the rear subframe. Upon rigid barrier impact, the driveshaft essentially ceases its forward motion while the rest of the vehicle continues to move forward. Since the driveshaft is connected directly to the RDU, the RDU also stops and impacts the subframe which still has forward motion. This impact causes the high deceleration pulse. Due to a lack of time during the summer internship, testing was not completed to study the interaction between the driveshaft and the RDU.

RDU COMPARATIVE ANALYSIS

Using CAE models, simulations were run to observe the movement of the RDU relative to the subframe, as well as to determine the stress concentrations experienced by the front and rear RDU mounting brackets. CAE renderings of the RDU on the undercarriage are shown in Figure 8 below. Figure 9 shows the undercarriage of the Edge with the driveshaft connected to the RDU mounted on the rear subframe. Figure 10, page 11, shows the stresses incurred in the rear RDU mounting brackets during an NCAP simulation. During impact, these brackets slightly deform as they impact the rear subframe. The deformation of the brackets observed corresponds to large load transfer due to the thickness of the metal.

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Figure 8. Assembly drawing of chassis sub-system

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Figure 9. CAE rendering of undercarriage of Edge showing RDU connected to driveshaft and rear subframe

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Figure 10. Stress concentrations observed in rear RDU mounting brackets during impact

The clearances between the driveshaft keyway brackets and the crash disks were measured and determined to be larger than the distance the RDU brackets allow the RDU to travel. This led to the conclusion that the clearances between the RDU and subframe are not large enough to prevent impact, which contribute to the high deceleration pulse. These clearances are shown in Figure 11 and Figure 12, page 12. 78.7 mm of clearance exists between the left rear side of the RDU and the rear subframe, and 47.7 mm of clearance exists between the center of the RDU and subframe. However, due to the rear RDU mounting brackets, this clearance is not fully utilized. Additional clearance from the driveshaft crash disks is obtained from the deformation of the part, but not enough to avoid the RDU impact. This poses an issue because the clearance needed to force the breakaway of the keyways in the driveshaft is greater than 17.0 mm. The crash disk impacts the bracket at 17.0 mm, but additional displacement is needed to completely disengage the keyways due to the deformation of the brackets. Therefore, the driveshaft does not separate from the undercarriage early enough during the impact to prevent the interaction between the rear subframe and the RDU. Our goal is to increase the clearance between the RDU and rear subframe by modifying the rear mounting brackets.

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Figure 11. Rendering of clearances measured between RDU and rear subframe (right rear RDU mounting bracket not shown)

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Figure 12. Rendering of clearance measured between crash disk and driveshaft breakaway bracket

During a hoist review, it was observed that the front casting of the RDU broke at the front RDU mounting brackets during impact, as shown in Figure 13 and Figure 14, page 13. As mentioned in the "Prior Work" section, this event was replicated with component-level testing, however, the interaction between the driveshaft and RDU was not replicated due to lack of time.

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Figure 13. CAE rendering of rear subframe showing first impact point of RDU casting

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Figure 14. Post-crash photo of broken RDU casting

SPECIFICATIONS

The RDU used on the Edge is used on several different vehicles, and therefore cannot be changed. Instead, the brackets attaching the RDU to the subframe will be modified to allow increased clearance between the RDU and the subframe, along with greater ease of longitudinal travel. Also, the distance between the crash disks and the drivetrain mounting brackets will remain 17.0 mm. This value is a Ford industry standard and cannot be changed.

The front mounting brackets prevent adequate travel of the RDU, and will therefore be weakened to allow movement of the RDU during an impact. In order to remain compatible with the unmodified RDU, it will retain the same connection points, general geometry, and a thickness of 3.6 mm. The relative locations of the bolt holes and bushing can be seen on the original bracket in Figure 15, page 14.

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Figure 15. Sketch of original front RDU bracket

The left rear bracket, when viewed from the top, will retain the two connection points to the RDU and remain 6 mm thick. A 12 mm diameter, 100 mm long rod will protrude from the bracket where the connection point to the subframe was previously located. The relative positions of the bolt holes along with the location of the subframe connection point can be seen in Figure 16, page 15.

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Figure 16. Sketch of original left rear RDU bracket

The right rear bracket, as viewed from the top, will retain the three connection points to the RDU and will be 5 mm thick. A 12 mm diameter, 100 mm long rod will protrude from the bracket where the connection point to the subframe was previously located. The relative positions of the bolt holes along with the location of the subframe connection point can be seen in Figure 17 below.

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Figure 17. Sketch of original right rear RDU bracket

The amount of material to be removed from the flange on the subframe should be as small as possible and still allow free movement of the castings on the front of the RDU as well as the front RDU brackets. The specific quantity of material removed is not a concern since this component is a non-load bearing piece.

CONCEPT GENERATION

Several brainstorming sessions were carried out with our group, Ford engineers, and Dr. Gregory Hulbert. Also, a functional decomposition was performed of the rear RDU to focus on functionality instead of form before brainstorming. A copy of the functional decomposition can be found in Appendix B. The concepts can be separated into several general categories:

- Modifying the brackets supporting the RDU
- Weaken the bolts attaching the RDU to brackets
- Reduce distance needed for driveshaft crash disks to break out keyways
- Redesign subframe
- Give RDU more longitudinal freedom

All of these concepts allow the RDU to travel the distance required for the crash disks to break away the keyways during an impact. A larger list of concepts can be found in Appendix C.

The first of the categories, weakening the brackets that support the RDU, would simply allow the RDU to travel more freely during an impact. The clearance necessary to allow the driveshaft crash disks to break out of the keyways is there, however, the mounting brackets of the RDU prevent that entire distance from being utilized. A modification like this could involve, for example, weakening the front mounting brackets, the teal and yellow areas on either side of the orange RDU in Figure 13, page 13.

The next concept category involves allowing the bolts joining the RDU to its mounting brackets to shear upon impact. These bolts are the teal bolts attached to the yellow brackets in Figure 18, below. If these bolts were to shear upon impact, the RDU would be free to move the full 47.7 mm before making contact with the subframe.

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Figure 18. Rendering of RDU bolts attachment to rear brackets

Our next concept was to redesign the crash disks on the driveshaft so that they required less travel before breaking out the keyways and letting the driveshaft drop out of the load path in the vehicle. Figure 19 below shows the crash disk (dark purple) and keyway (red). By reducing this distance, clearances between the RDU and subframe would no longer be an issue, and remain unchanged.

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Figure 19. Rendering of crash disk and driveshaft breakaway bracket

Another concept was to make design changes to the subframe. This could be done either by redesigning the mounting points or cutting out parts of the subframe. Redesigning the mounting points in such a way allows more movement in the system to increase the travel of the RDU. Also, cutting out pieces of the subframe would allow more clearance for the RDU to move in an impact before contacting the subframe itself.

Our final concept involves the idea of giving the RDU increased movement in the longitudinal direction, while keeping it fully supported in the vertical direction. This can be accomplished by shortening the two rear mounting brackets of the RDU and using longer bolts, which will run through the bushings on the rear of the subframe. This would give the RDU freedom to move longitudinally during impact, fully utilizing the clearance present.

CONCEPT SELECTION

As listed in the "Concept Generation" section, we had five major concept categories. We first decided which category was most feasible for our redesign, and then pursued more detailed designs within that category. The specific design variations within this category are compared in the Pugh Chart in Appendix D.

The first category we dismissed was reducing the distance needed for the driveshaft crash disks to break out the keyways. The advantage of this is that it would not require any modifications to the RDU or the subframe where the RDU attaches. However, implementing this option is not possible due to a Ford industry standard that requires 17 mm of travel before the crash disks begin to break out the keyways.

The second category, redesigning the subframe, has an advantage in its simplicity; it only requires that material be removed wherever interference occurs between the RDU and the subframe. However, implementing this option is not feasible due to the complexity of the subframe. Many components of the vehicle interact with and depend on the subframe, and any modifications that could possibly interfere with its structural integrity would involve analyses far outside of the scope of this project.

Allowing the RDU to have unrestricted movement in the longitudinal direction was also considered, as it would eliminate the problem of the RDU not being able to take advantage of its entire clearance before impacting the subframe. In normal vehicle operation, though, this option may not provide the RDU with the support and durability it needs in the longitudinal direction.

The next concept, weakening the bolts connecting the RDU to its mounting brackets, was considered in the alpha design since it would allow the RDU to travel during impact, allowing it to utilize its full clearance before impacting the subframe. However, a new problem arises in that the brackets themselves would now become an object of interference between the RDU and the subframe. Even though the RDU is free to move once the bolts have failed, the brackets will still be in between the RDU and subframe, causing an impact earlier than desired.

The final concept, modifying the brackets that support the RDU was also considered for the alpha design. This would involve weakening the front brackets, allowing the RDU to move freely during an impact, and shortening the rear brackets and using longer bolts to connect them to the subframe so that the brackets will not be in the way of the RDU as it travels in an impact. A problem with this concept, though, is that the longer bolts required to use the shorter rear brackets could cause problems with vertical stiffness of the RDU/subframe assembly.

CONCEPT DESCRIPTION

The concept that will provide a proof of concept includes two main elements. The first main element is the rear right RDU bracket (Figure 20 through Figure 22, page 18). The rear right bracket utilizes the same two point attachment to the RDU. The customer utilizes this RDU on several of its all wheel drive vehicles, so it imporant to the customer not to change any of the attachment points to the RDU.

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Figure 20. Perspective view of the rear right RDU bracket

As shown in above, the bracket includes a cylindrical rod extending from a distal end of the rear left bracket. The vertical and horizontal position of the rod corresponds with the location of the bolt that extends through the current rear right RDU bracket into the subframe. However, the bracket was shortened to increase the clearance between the face of the bracket that the rod is attached to and the rear subframe.

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Figure 21. Second perspective view of the rear right RDU bracket

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Figure 22. Third perspective view of the rear right RDU bracket

The second main element is the rear left RDU bracket (Figure 23 through Figure 25, pages 20-21). The rear left bracket utilizes the same three point attachment to the RDU to meet the customer needs. Similar to the right rear RDU bracket, the left rear RDU bracket utilizes a similar rod extending from a surface facing the subframe. The rear left RDU bracket is also shortened to increase the clearance between the bracket and the subframe.

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Figure 23. Perspective view of the rear left RDU bracket

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Figure 24. Second perspective view of the rear left RDU bracket

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Figure 25. Third perspective view of the rear left RDU bracket

Figure 26 below shows the concept left and right rear brackets attached to a subframe on the test vehicle. As shown in the figure, the clearance between the brackets and the subframe is greatly increased. The concept brackets will allow for more along the longitudinal direction while maintaining the RDU in the proper vertical alignment.

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Figure 26. Bottom view of rear RDU brackets on the test vehicle

PARAMETER ANALYSIS

To determine the specific parameters of the final design, several analyses were conducted. Initially, evaluations began with performing a CAE analysis to determine the root cause of the deceleration pulses experienced during NCAP testing. Video analysis of the test initially led us to believe the pulses were caused by the fact that the engine roll restrictor remained stationary during impact. Further NCAP testing proved that the root cause of the pulses was not the engine roll restrictor, but may be the interaction between the RDU brackets and the subframe. After this discovery, the focus of our project was shifted towards the interaction of the RDU brackets and the subframe.

After analyzing the NCAP testing results, a CAE analysis was conducted that allowed specific interactions between vehicle components to be identified and evaluated. The model showed that there were stresses exhibited during the interaction of the RDU brackets and the subframe as seen in Figure 10, page 11. This analysis further led the team to consider the interaction between the RDU brackets and the subframe as the model was run with and without the brackets to see difference in pulses between the two. The model showed that a decrease of between 9 and 15 G's could be achieved with the removal of the brackets. Since the removal of the brackets was not a feasible option, we conducted further analysis to see how the destructive interaction could be remedied.

Further analysis was conducted through component-level testing. The component-level testing replicated the impact observed during an NCAP test. Video analysis during the testing allowed the fracture methods to be analyzed as well as to observe the deflection experienced between the RDU and the rear subframe. From this analysis, we determined that the clearance between the RDU brackets and the rear subframe were not being utilized because of the geometry of the brackets.

To reduce the interaction between the RDU brackets and the subframe, we analyzed how the load transferred during a crash and the timing of the interactions. The NCAP test results showed that the timeframe within which the RDU brackets and the subframe interact is where the pulses spike. We investigated how to increase the time of the interaction so that the driveshaft breaks away and disrupts the load path. To accomplish this, we suggested that the clearance between the RDU and the subframe be utilized so that there is more distance required to travel before the brackets can hit the subframe. To do this, we proposed shortening the brackets and attaching rods which connect to the brackets and the subframe so that the RDU is free to move in the longitudinal direction while still being supported in the vertical direction.

The material of the brackets was not changed during our project because in doing so, we would have to investigate the loads experienced during normal operation which was unavailable to us in the timeframe of our project. In addition, the general shape of the bracket, including the specifications previously mentioned in "Specifications" was preserved. The short timeframe limited the team in terms of providing a detailed design of our concept before fabrication. The bracket was fabricated off-site while CAD drawings were constructed so that the vehicle could be crashed in a timely fashion. Because the bracket was fabricated off-site, the specific dimensions of the rod as well as the point where the bracket was shortened was not determined by the team. Rather, we presented our concept and Ford created the prototype.

Several analyses were conducted to determine further information about our part and how we can improve it in terms of the material, assembly, environment, safety, and manufacturing process. A detailed explanation of these analyses can be found in Appendix I.

While the material for our current prototype could not be changed due to our timeframe, a detailed analysis was conducted using the Cambridge Engineering Selector (CES) software. Only one major component was involved in our assembly, the bracket, and after discussing with our team leader, Dr. Gregory Hulbert, the analyses were only carried out on this part. Inputting our function, objectives, and constraints into the program, several material options were delivered. We narrowed down our top five choices and the final choice to manufacturing the redesigned brackets is a low alloy steel, AISI 4620.

We determined the assembly efficiency of our design for the left and right rear brackets. We integrated the rod and bracket decrease the assembly time and increase the assembly efficiencies, which are summarized in Table 1 below.

	Original Design Assembly Efficiency	**Redesign Assembly Efficiency**
Rear Left Bracket	0.188	0.257
Rear Right Bracket	0.235	0.321

Table 1. Assembly Efficiencies for Original Design and Redesigned Brackets

The best materials for our part were analyzed for environmental sustainability using SimaPro. After conducting an in-depth analysis comparing the low alloy steel, AISI 4620, and wrought aluminum alloy, 2026-T3511, we determined that the low alloy steel is better for the environment. We further investigated the effects of materials on the environment by comparing the low alloy steel, AISI 4620 to carbon steels, AISI 1022, AISI 1030, and AISI 1118. The results of the analysis showed the carbon steel to be better for the environment than the low alloy steel. Hence, we suggest that Ford investigate the mechanical properties of carbon steel to see if it will maintain the required specifications of the bracket.

A detailed risk assessment was conducted for our assembly to determine potential hazards installers might face during installation. A copy of the DesignSafe report can be found in I4, page I16. Most of the risks encountered during installation were mechanical while some were related to ergonomics and human factors. After conducting the risk assessment, we lowered all the risk levels to low.

Finally, we determined the best manufacturing process for a production volume of 150,000 parts for each left and right rear bracket. Using CES, we determined that die casting would be the best manufacturing process. This process would also allow us to integrate the bracket and the rod, thus increasing our assembly efficiency as previously mentioned.

FINAL DESIGN

Engineering drawings of the final design can be seen in Figure 27 through Figure 32, pages 24-25, and in Appendix E. The attachment points to the RDU are the same as the stock brackets on the Ford Edge. The brackets were shortened to increase the clearance between the RDU and the subframe. In addition, a separate a separate nut will be added to the end of the rod to prevent the rod from moving out of the bushing during testing.

Part No.	Part Description
1	Left rear RDU bracket
2	Right rear RDU bracket

Table 2. Parts list of major components

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Figure 27. Left rear bracket side view engineering drawing

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Figure 28. Left rear bracket top view engineering drawing

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Figure 29. Left rear bracket bottom view engineering drawing

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Figure 30. Right rear bracket front view engineering drawing

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Figure 31. Right rear bracket bottom view engineering drawing

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Figure 32. Right rear bracket side view engineering drawing

The final design will be able to validate the root cause of the large pulses observed during frontal impact crash by crashing a vehicle with the new brackets. We will not be able to evaluate the interaction of the new brackets on other systems on the vehicle as well as its effects on vehicle dynamics. Further explanation of the prototype validation will be addressed later in the report.

FABRICATION PLAN

All modifications to the test vehicle and RDU mounting brackets were completed at the Livernois Vehicle Development Center. Sections of the front RDU mounting brackets and subframe flange were removed to encourage fracture or deformation and to reduce impacts between the RDU and subframe, respectively. In prior testing, the casting of the RDU connected to the front brackets often fractures, so the removal of the supporting flange and notching encourages fracture in the brackets to reduce the impact between the front brackets and the subframe. The flange removal on the subframe was performed to remove the possible impact between the brackets and subframe. The subframe was welded closed where material was removed because the purpose of the flange was to contain the original welding. Photographs of the brackets and subframe before and after modifications are shown in Figure 33 through Figure 35, page 27. In Figure 33, the areas marked in yellow were removed on the brackets and subframe. Figure 34 shows the final alterations to the now yellow brackets and beige subframe from the front view and Figure 35 shows the modifications from the rear view. Figure 36, page 28, shows the original engineering drawings of the front bracket with the material removed colored in red.

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Figure 33. Pre-modification photo of subframe. Material removed is circled and marked in yellow on the brackets and the flange on the subframe.

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Figure 34. Final modifications to front RDU brackets and subframe circled or pointed out in red.

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Figure 35. Final modifications to the front RDU mounting brackets as seen from the rear of the vehicle.

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Figure 36. Right-hand and isometric views of front bracket with material removed colored in red.

The modifications to the rear RDU mounting brackets are shown in Figure 37 and Figure 38, page 29, and Figure 39, page 29. These changes involved cutting off the original end of the bracket that connects to the bushing in the rear subframe and welding part of the original bracket closer to the RDU to utilize as much clearance between the end of the bracket and the rear subframe. A metal rod was welded onto the end of the bracket to slide through the bushing, providing only vertical support to the RDU from the rear. Figure 37 is a CAE rendering of the original rear RDU mounting brackets, shown in yellow. Figure 38 contains engineering drawings of the right and left rear mounting brackets with the removed material shown in dashed lines. Figure 39 is an actual pre-test photograph of the RDU as seen from the top. Not shown are the washers welded to the ends of the rods to prevent the rods from sliding all of the way through the bushings due to the forward movement of the RDU during impact.

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Figure 37. CAE rendering of baseline mounting brackets.

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Figure 38. Front engineering drawings of right and left rear brackets. Material removed is between red dashed lines.

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Figure 39. Top view of final modifications with rods welded to rear brackets.

Since all of these modifications were performed by skilled technicians, we do not know the details of the modifications. If this design were production ready, due to the production volume of parts, these brackets would be manufacturing using a die casting process. This manufacturing process was determined using CES software, and a more detailed explanation can be found in Appendix I.

VALIDATION RESULTS

The focus of our project was two-fold: first, to provide a "proof-of-problem" that the impacts between the rear RDU mounting brackets and subframe is a major contributor to the drivetrain-related deceleration pulse and to determine an alpha design for new rear mounting brackets.

PROOF OF PROBLEM The problem was verified via an NCAP test at the Crash Barrier Building. CAE models have been run with alterations made to the RDU brackets and subframe, which have virtually confirmed through simulations that physical changes to the rear subframe assembly will reduce the deceleration pulse, which gave us the confidence to pursue making these modifications to a full vehicle. The model provided us with a range for the ideal decrease in the pulse, which allowed us to compare the success of our preliminary design changes to a recent NCAP test of a Lincoln MKX, which experienced a peak pulse of 62 G's. A simulation was also run without a driveshaft to determine the best pulse reduction possible, which was 9 G's.

In order to run an NCAP test, much preparation needed to occur. First, the vehicle was ordered and delivered to the Livernois Vehicle Development center in Inkster, MI, and a test planning form was completed. This test planning form included information such as test instrumentation and cameras, as well as the modifications made to the vehicle to update it to a 2009 Lincoln MKX and to incorporate our design changes to the RDU brackets. For our specific test, we requested numerous accelerometers, contact switches, and break-away switches to capture as many events as possible during the crash. The accelerometer located at the rocker at the B-pillar was the channel compared to past crashes to determine if our modifications were successful.

A variety of camera views were used to visually analyze the physical implications of the modifications to the RDU brackets and subframe. These camera views included the standard pit views, taken from the floor underneath the impact zone through glass windows, and new camera views were developed to capture the events from different angles. These additional cameras included two attached to the undercarriage where the fuel tank was removed to record the movement of the RDU and brackets from the front view, and two more cameras were mounted inside of the vehicle. A hole was cut in the floor pan in the rear of the vehicle so that these two additional cameras could capture the impact from above the RDU and subframe and were specifically pointed to see the interaction between the modified rear RDU mounting brackets and rear subframe.

Using the data from the instrumentation, video analysis, and a hoist review to look for witness marks and interactions between the RDU and rear subframe, we were able to determine if our modifications were successful in decreasing the deceleration pulse due to the drivetrain.

FINAL DESIGN PROTOTYPE Once it was determined that by removing the interactions between the RDU and rear subframe decreases the drivetrain-related pulse, we developed an alpha prototype design. The focus of this redesign centered around increasing the clearance between the RDU and subframe by providing more lateral movement of the RDU before it or its mounting brackets impact the subframe. After brainstorming, numerous designs were sketched out and presented to both safety and durability engineers to determine which design was most feasible for both crashworthiness and durability standpoints.

The safety engineers asked us to refine the design we had proposed for the full-vehicle test. After speaking to durability engineers, we learned that our design was not production ready due to NVH issues. By removing the direct attachment of the bracket to the subframe through the bushing, we introduced a cantilever motion that would not withstand durability and vehicle dynamics testing. The brackets were designed to be so thick and stiff to decrease NVH problems. It was suggested that we investigate a more durable design, but due to a lack of time, we were unable to complete this analysis.

NCAP CRASH TEST RESULTS After completing the post-test analysis of our crash, we determined that our prototype RDU bracket redesigns were a success. We exceeded our expectations of a 9-15 G drop by experiencing a decrease of 26 G's from the latest baseline NCAP test. The deceleration pulse due to the drivetrain in our test was 36 G's, which is much lower than the 62 G's experienced by the prior run. Video analysis confirmed our original belief that the impact between the RDU and subframe were substantial, as observed by the longitudinal shift of the RDU, but with the new brackets, the impact was much less severe, and now occurred between the actual RDU and subframe instead of the brackets.

Break-away switches were instrumented on the keyways on the driveshaft which recorded the disengagement of the brackets 3 ms before the peak pulse, which means the driveshaft decoupled before the main drivetrain-related pulse instead of during or after, which led to the higher pulses previously experienced. The decoupling of the driveshaft before the peak pulse prevented some of the load transfer through the RDU to the subframe.

Figure 40, page 32, shows the deceleration plot overlay of the baseline AWD in blue and the modified AWD in red. As shown in the plot, the curves are very close leading up to the drivetrain-related pulse (shown in the green box), and the peaks occur within 1 ms of each other, verifying that the pulses can be compared as both resulting from the interactions of the drivetrain. The subsequent peaks are due to the impacts of the back-up structure, which is not considered part of the drivetrain.

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Figure 40. Deceleration pulse overlay of baseline and modified AWD vehicles.

Figure 41 through Figure 43, page 32, are post-test pictures of the vehicle after impact. Figure 41 shows the rear view of the RDU and subframe. The rods have pushed through the bushings as expected. Figure 42 shows the front RDU mounting brackets, which stayed intact although they were notched and weakened to encourage fracture. Figure 43 shows the decoupled driveshaft, which disengaged prior to the drivetrain-related pulse.

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Figure 41. Post-test photograph of rear subframe with rods through bushings

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Figure 42. Post-test photograph of front RDU mounting brackets

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Figure 43. Post-test photograph of decoupled driveshaft

Post-processing of the pulse data was performed in MADYMO (Mathematical Dynamic Modeling), which predicts the impact of the crash pulse on an occupant. When comparing our modified AWD to the previous AWD NCAP crash, the chest G's experienced by the driver are decreased by 3%, and those experienced by the passenger are decreased by 8%. These values further emphasize the improvement in the deceleration pulse to both the vehicle and the occupant.

DESIGN CRITIQUE

If this project were to be repeated, a vastly different path to the final product would have been taken. With the knowledge gained through the project, we would have known to begin with a focus on redesigning the RDU supports. While the work done with the roll restrictor was not insignificant in that it helped us to realize that it was not the root cause of the deceleration pulses, it did consume a significant amount of time that could have been better spent. Component-level testing similar to that done with the roll restrictor would have been carried out with the driveshaft, RDU, and rear subframe in order to gain a more detailed knowledge of the interaction between these components during an impact. Also, detailed analysis of the rear brackets, both the original and redesigned, would have been carried out. This would have included at the least a finite element analysis of the components to determine the stresses they must endure during both normal operation and impact, and to predict the fatigue life of the redesigned component.

Our design, in theory, works very well in that it accomplishes the goal simply. The main goal of the redesign was to allow enough clearance between the RDU and rear subframe so that the driveshaft keyways could break away and stop the load transfer before an interaction between the RDU and rear subframe occurs. The redesigned part is still able to provide the RDU support in the vertical direction and prevent it from rotating relative to the subframe, and it also gives the RDU nearly complete longitudinal freedom. Weakening of the front RDU brackets and removal of material on a flange attached to the subframe, along with the modified rear brackets, allows the RDU to take full advantage of the clearance already between it and the rear subframe.

While this redesign solves the initial problem, giving the RDU longitudinal freedom to move could potentially introduce an entirely new set of problems that would need to be addressed before this design were implemented into a production line. Flanges were removed from the front RDU brackets so that, during impact, the front brackets would release the RDU and allow it to travel freely. Those flanges were a part of the original brackets' design to increase rigidity, and without them, the bracket becomes more susceptible to deformation, and therefore higher stresses and fatigue. Also, the rear brackets were previously tightly clamped to the rear subframe, nearly stopping motion in all directions. Without being clamped, the new design becomes vulnerable to increased vibration, deformation due to cantilever motion, and fatigue.

RECOMMENDATIONS

As a lack of time was a large factor in our project, many of the analyses we would have liked to perform were not completed. We have proved that by increasing the clearances between the RDU brackets and rear subframe, the drivetrain-related deceleration pulse can be greatly reduced. The design we proposed and tested is adequate from a crash safety standpoint, but this redesign opens up issues for many other functional teams, including NVH (Noise, Vibration, and Harshness), vehicle dynamics, durability, and driveline performance. We suggest presenting our design and research to the current RDU bracket design and release (D&R) engineer as a proof of concept that needs further development to design more feasible brackets to benefit crashworthiness.

Many of our potential design changes were presented in our "Concept Generation" section, and we suggest looking into material changes to a less stiff material. We also feel that a new geometry for the brackets may be beneficial if the brackets can be designed to collapse or bend to utilize all of the available clearance space between the RDU and rear subframe. We recommend investigating the possibility of modifications to the rear subframe to provide more clearance for the brackets.

CONCLUSIONS

Although our project focus changed halfway through the allotted time, we were able to build off of our knowledge gained from studying the crash pulse and the roll restrictor and apply it to the rear differential unit (RDU). Through hoist reviews, CAE analysis, and dimensional analysis, we determined that the impacts between the RDU and rear subframe substantially contribute to the high deceleration pulse related to the drivetrain. We provided a proof of concept and basic alpha design that successfully reduced the severity of the drivetrain-related pulse, which was demonstrated with an A-to-B comparison NCAP full vehicle test.

Although we have resolved the high pulse issue from a crash safety standpoint, the current design is not production-ready due to issues with durability and NVH, so more developmental work must occur to refine the RDU mounting brackets for future vehicles.

ACKNOWLEDGEMENTS

The team would like to thank Mr. Steve Kozak for his support and sponsorship of this project, as well as giving us access to so many resources through Ford Motor Company. We also could not have completed this project successfully without the support and guidance from Dr. Gregory Hulbert. We greatly appreciate the efforts of Mr. Kirk Arthurs in lending engineering support, and the authors would like to recognize the time and dedication of Mr. John Fazio and his help with CAE simulations, component- and full-vehicle testing, and overall engineering support. We would also like to thank the engineers and technicians at the Livernois Vehicle Development Center and Ford Crash Barrier Building for their work in preparing the vehicle for crash.

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APPENDIX A. DESIGN REVIEW #2

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EXECUTIVE SUMMARY

During a full-frontal impact into a rigid barrier at 35 mph, the Ford Edge and Lincoln MKX experienced high G-force deceleration pulses in excess of 60 G's. The average deceleration pulses in other vehicles range from 30-40 G's. It was determined that the engine roll restrictor was not deforming during impact and thus acted as a direct load path, causing these pulses. The goal of this project is to redesign the roll restrictor to deform or decouple during impact in order to reduce the drive train deceleration pulse in the vehicle.

This project has an accelerated timeline due to the prototype manufacturing time. Literature analyses, benchmarking, and meetings with engineers at Ford were conducted to determine what types of engine roll restrictors are on the market and to determine the requirements and constraints of the redesigned part. The project has been allotted a budget of $100,000 to develop and implement the prototype through a full frontal impact on a Ford Edge.

The redesigned roll restrictor must be able to withstand loads of 6800 lbf in tension and 3400 lbf in compression, and have a natural frequency of at least 500 Hz. It must maintain the current 3-point transmission case attachment and meet current packaging restrictions. Finally, it will have a predetermined failure mode so that it will deform or decouple during impact.

Component-level testing has been completed at the Safety Innovation Lab in Dearborn. This testing provided an opportunity to gain baseline force data on the original design, as well as data on modified restrictors to develop an alpha prototype design. We are awaiting results from a New Car Assessment Program (NCAP) computer aided engineering (CAE) model to determine the forces that the restrictor experiences during impact, which will affect the final design. Once the final design has been determined, CAD models will be completed and finite element analysis (FEA) results will be generated to confirm the design and its ability to withstand current durability standards.

Due to authorization and confidentiality issues, most of the work must be done in conjunction with Ford Motor Company engineers, including CAE and FEA at the Dearborn facilities. Also, since we do not have the capabilities to do so, final prototyping will be carried out by the Ford Motor Company.

INTRODUCTION

The National Highway Traffic Safety Association (NHTSA) rigorously tests new vehicles to ensure their crashworthiness and assign star ratings. NHTSA's NCAP involves a full-frontal impact into a rigid barrier at 35 mph, which simulates a 70 mph head-on crash between two vehicles.

During a summer internship, one of the team members was asked to investigate the root cause of the high G-force deceleration pulses experienced by the Ford Edge and Lincoln MKX. The pulses are often caused by internal impacts within the vehicle as components buckle and crush, and these pulses are detected by accelerometers placed throughout the vehicle structure. High deceleration pulses in the body frame lead to higher loads experienced by the passengers in the vehicle. The main focus in the Ford Edge investigation was concentrated on the pulse related to the drivetrain, which includes the power take-off unit, drive shaft, engine roll restrictor, and rear differential unit. This high pulse was measured at the rocker at the B-pillar, which is located below and behind the driver's side door. These major components are shown on Figure A1, below.

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Figure A1. Undercarriage view of Ford Edge, post-crash.

It was determined that the engine roll restrictor was acting as a direct load path during impact, bottoming out between the transmission, power take-off, and drive shaft. Furthermore, the roll restrictor does not deform during the impact; it remains stationary until it fractures the transmission casing. These observations as well as the recommendation that the roll restrictor be redesigned to disrupt the direct transfer of the forces through the subframe of the vehicle were presented to Steve Kozak, the Chief Safety Engineer at Ford Motor Company. Mr. Kozak then requested that the work started over the summer be integrated into a senior design project for the University of Michigan.

The roll restrictor is used in the vehicle to reduce the vibrational motion of the engine while keeping it in alignment and connected to the subframe. It consists of three components: the link, the adjoining bracket, and the plate as seen in Figure A2 below. The link is mainly aluminum and contains a rubber damper that decreases noise and vibrations in the system and connects to the subframe. The bracket, made of cast iron, is attached to the transmission case and will be the focus of the redesign. An aluminum plate holds the entire assembly together and serves as an attachment point for the heat shield. Figure A3 below shows a close-up view of the roll restrictor connected to the transmission and subframe with the heat shield attached from a bottom and side view.

Proprietary Information

Figure A2. Roll restrictor

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Figure A3. Bottom (left) and side (right) views of roll restrictor attached to vehicle underbody.

The goal of this project is to redesign the bracket so that the roll restrictor deforms or decouples, therefore disrupting the load path and ultimately reducing the deceleration pulses due to the effect of the drivetrain. Our team will be working with the design and release (D&R) engineers at Ford to redesign the bracket to withstand the loads experienced by the vehicle during normal driving conditions and fail during a high speed frontal impact. Due to the machining of the component, prototyping will be completed in Ford's Dearborn facilities.

Based on meetings with the D&R engineers at Ford, the major design specifications for the redesign are as follows:

1. Load carried in tension: 6800 lbf
2. Load carried in compression: 3400 lbf
3. Resonant frequency: 500+ Hz
4. Maintain 3-point attachment to transmission case
5. Maintain major geometry of link (platform component for other vehicle programs)
6. Withstand current durability requirements

If we can meet all of these requirements, the prototype part will be tested on at least one full vehicle at the Crash Barrier Building in the Dearborn Development Grounds in April. We have already obtained an Edge from the same model year as the investigated program, so it will be a direct comparison based on the changes made to the roll restrictor, as the current models have already introduced changes in the driveshaft. The results will confirm our design changes if we can lower the G-force deceleration to a level similar to those found from the CAE model run without a roll restrictor present. This predicts the best case scenario for the decrease in the G-force deceleration pulses due to the effects of the roll restrictor.

ENGINEERING SPECIFICATIONS

SELECTION OF ENGINEERING TARGETS

An initial meeting was held with Paul Jabbour, a Ford D&R engineer, in early January to determine the customer requirements for the redesigned roll restrictor. The redesigned part requires a 3-point attachment to the transmission case while maintaining a similar size to allow for cross compatibility between the various vehicles on which it is currently used. It should be able to support the load cycles observed during the life of the vehicle. Furthermore, the key challenge to be addressed by this project is to reduce the deceleration pulses observed during NCAP testing.

During further communication with Mr. Jabbour in mid February, it was noted that the redesigned part must not experience loads during normal operation that exceed the yield strength (54 kpsi for cast iron) of the material as determined by a finite element analysis (FEA).

In order to determine the engineering goals of this project, a quality function deployment (QFD) chart was used to relate the customer requirements to the engineering goals. The QFD chart for this project can be seen in Figure A4, page A6.

	Customer Weights	Material	Load carried in tension (lbf)	Load carried in compression (lbf)	Resonant frequency (Hz)	Breaks at a load of (lbf)	Points of failure (#)	Height (in)	Width (in)	Depth (in)
Material										
Load carried in tension (lbf)		9								
Load carried in compression (lbf)		9	1							
Resonant frequency (Hz)		3								
Breaks at load of (lbf)		3								
Points of failure (#)						9				
Height (in)			1	1	3					
Width (in)			1	1	3					
Depth (in)			1	1	3					
Maximum Stress (kpsi)		3				3	1			

		Technical Requirements										Benchmarks
Customer Needs	Customer Weights	Material	Load carried in tension (lbf)	Load carried in compression (lbf)	Resonant frequency (Hz)	Breaks at a load of (lbf)	Points of failure (#)	Height (in)	Width (in)	Depth (in)	Maximum Stress (kpsi)	*Current model in Ford vehicles*
3 point attachment to transmission case	9						3	1	1	1		4
Fails during frontal impact	2	9				9					3	1
"Infinite" lifetime	6	9	1	1	1							3
Minimize weight	4	9						9	9	9		2
Supports engine	11		9	9	1						1	4
Fast prototype	1	9						1	1	1		2
Fits in space provided	10		1	1	1			9	9	9		4
Cross compatible between vehicles	8							3	3	3		4
Doesn't break transmission case	3	3				3					3	1
Reduces deceleration pulses	6					9	9				3	1
Passes FEA	7	3	3	3		1					9	4
Importance		147	136	136	27	88	81	160	160	160	107	
Target values		Fe/Al	6800	3400	500+	5300	2	3	5.4	3.5	54	
Current Model		Fe/Al	6800	3400	500+	N/A	0	3	5.4	3.9	54	

Figure A4. QFD chart for engine roll restrictor redesign project

The relative importance of each technical requirement was determined by multiplying its correlation value to a particular customer need by the weight of that particular need and then summing these values for each customer need. A larger customer weight indicates a stronger customer desire. Table 1, page A7, lists each technical requirement and its relative importance. Due to constraints set forth by Ford, the minimum loads that the part can withstand, as well as the natural frequency of the part cannot be changed. It is the goal of this project, therefore, to concentrate on the specifications with the highest values of importance: the material of the part, its height, length, and width. These specifications should be modified to meet the customer's needs. The redesigned part will be manufactured from both cast iron and aluminum like the original to reduce the number of changes to the manufacturing process, especially since the link is used on multiple Ford vehicles. Also, by modifying the dimensions and other geometrical features of the bracket, not only can weight be reduced, but prescribed failure points can be introduced into the design. Through predicting the part's failure, deceleration pulses during a 35 mph frontal impact should be reduced.

Technical requirement	Target value	Relative importance
Material	Al	156
Load carried in tension (lbf)	6800	136
Load carried in compression (lbf)	3400	136
Resonant frequency (Hz)	500+	27
Breaks at a load of (lbf)	5300	70
Points of failure (#)	2	63
Height (in)	3.0	169
Width (in)	5.4	169
Depth (in)	3.5	169
Maximum Stress (kpsi)	54	101

Table 1. Technical requirements and their respective target values and relative importance

Component-level testing was carried out at the Ford Safety Labs in order to determine the loads at which failure of the link or bracket occurs. It was determined that the bracket will not break during impact; the bolts will shear or other parts of the system will fail first. Several modified brackets were tested until a modification was found that caused the bracket to fail in a consistent manner during impact. FEA will be performed with the help of Ford engineers to determine if this bracket modification can withstand the loads that will be placed upon it in normal use without exceeding the maximum stress (for cast iron) of 54 kpsi and to ensure its resonant frequency is greater than 500 Hz. A computer aided engineering (CAE) model will be run to determine the forces that the roll restrictor will endure during normal operation, and FEA will again be used to determine if the part will withstand those forces over the lifetime number of cycles without fatiguing.

CONCEPT GENERATION

Several brainstorming sessions were carried out with our group, as well as with groups of mechanical engineering students who were unfamiliar with the project and could therefore provide unbiased insights into the design of the roll restrictor bracket. Also, the group performed a functional decomposition of the roll restrictor to focus on the functionality of the roll restrictor instead of the form before brainstorming. A copy of the functional decomposition can be found in Appendix A. The concepts can be separated into several general categories:

- Remove material from specific parts of assembly to introduce failure modes
- Allow parts to fold like an accordion
- Change the material for one or more of the parts
- Change the orientation of the link
- Active mechanism to destroy/move part of the assembly

A full list of concepts and short descriptions can be found in Appendix B.

The first of the categories, material removal, involves making one or more modifications to the original design of the roll restrictor assembly. This can include general material removal, perforation of weak points, adding keyways that allow bolts to escape on impact, and machining crumple zones that would absorb energy during impact. Figure A5, page A8, shows a part modified in this way. A keyway has been machined into the plate so that the bolt supporting the link can slide out during impact. The bracket has undergone significant material removal in this case. Several ribs have been removed, the thickest cross section has been reduced, and one of the connection points to the transmission case has been removed. All of these modifications focus the load at the connection point of the link and the bracket to introduce failure during impact to promote the link to separate from the bracket and disrupt the load path.

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Figure A5. Concept design utilized material removal.

The next concept category involves changing one or more parts of the assembly so that they will buckle under the load of impact and absorb some of the impact energy before transmitting it. An example of this is shown in Figure A6, page A9. In this case, material is removed in the body of the link so that it is made up of four individual columns instead of two flat pieces. On impact, these columns will buckle, causing the link to collapse on itself. This deformation will absorb some of the energy during impact before transmitting the load further back in the subframe.

Proprietary Information

Figure A6. Crumple zone added in link. Material removed marked by red areas (black areas in b&w).

Our next concept category was to replace one or more parts of the assembly with a different material. One option in this category was to replace the entire bracket with aluminum. Doing this would greatly reduce the weight of the part, and the lower yield strength of the aluminum would allow the bracket to fail at a lower load than a cast iron bracket of the same geometry. However, because of the lower yield strength, the geometry of bracket would likely need to be changed significantly to reduce the stresses found in the bracket during normal operation to levels acceptable for aluminum. This could prevent the design from meeting the size specification required by the customer.

Another concept was to vary the angle between the link and the horizontal. This could be done in several ways including moving its mounting position on the bracket or through modifications to the subframe. When the link is mounted horizontally, an impact simply compresses the link and it does not move. If it were at an angle, an impact would exert a moment on the link as well as compress it. This moment will cause the link to rotate, thus absorbing some of the impact with the motion before it begins to transmit load to the subframe.

Our final concept utilizes an active mechanism that can detect an impact and cause the roll restrictor assembly to deform and absorb the impact energy, or can move the assembly out of the load path so that the energy is not transmitted.

CONCEPT SELECTION PROCESS

CONCEPT ADVANTAGES AND DISADVANTAGES

As listed in the previous section, we had five major concept categories. We first decided which category was most feasible for our redesign, and then pursued more detailed designs within that category.

The first category we dismissed was creating an active mechanism to destroy or move part of the assembly. The advantages of this is that it would guarantee that the roll restrictor would not transfer the load if the part was destroyed during impact, but this design path may be expensive and a malfunction in the sensor equipment would negate the positive effects.

The second category, allowing parts to fold like an accordion, has many advantages with the notching process as we can create a predictable fracture pattern, and we can also incorporate a weight reduction. This may cause durability issues by creating sharp notches which are stress risers, therefore this concept was not chosen.

Changing the angle of the link was carefully considered, as it would still be able to perform the function of the component and most, if not all, of the original roll restrictor could be used. However, this change alone is not aggressive enough to solve the problem; several interference issues could arise, prompting the redesign of several components.

Varying material is currently dependent upon the outcome of our FEA. Our top choice is aluminum, as it is a strong, light-weight material, but the disadvantages are the material cost and the lower material strength than cast iron. We are also unable to conduct component-level testing on an aluminum bracket at this time, which leaves us unsure of natural fracture points in the geometry.

The final category, removing material from specific parts of the assembly, was the main focus for our alpha design. The advantages behind this include cost and weight savings, encouraging fracture or failure at specific points, and maintaining all of the engineering specifications as listed by the D&R. The disadvantages include retooling costs and durability issues. The specific design variations within this category, along with an all aluminum design, are compared in the Pugh Chart in Appendix C.

COMPONENT LEVEL TESTING

The current concept was selected based on the results of component-level testing completed in Ford's Safety Innovation Lab (SIL) in Dearborn. Based on the crash analysis completed during the summer of 2007, the roll restrictor does not fracture or fail during an NCAP frontal impact, but instead breaks the transmission casing and acts as a load path through drivetrain. The purpose of the component-level testing was to isolate the restrictor and determine its natural weak points using a Bendix linear impact machine.

The Bendix test stand uses compressed nitrogen gas to push a carriage along a linear path to impact a component that is mounted to a stationary anvil. The carriage weight for our testing was set at 422 lbs on a two rod system with a flat-plate impactor. A fixture was built to replicate the subframe and transmission that surrounds the roll restrictor and was designed to only have one degree of freedom along the axis of the link. A load cell was attached behind the roll restrictor on the anvil to measure forces in the x, y, and z directions, as well as moments in the x and y directions. We are mainly concerned with the force in the z-direction, which is the axial direction of the impact. An accelerometer and linear variable displacement transducer (LVDT) were connected to the carriage for additional measurements. The Bendix test setup is shown in Figure A7, page A11. Photographs of the actual test fixture were not available at the time of this report.

Proprietary Information

Figure A7. Bendix Test Setup

The initial plan of action was to run baseline tests to determine the natural fracture pattern of the bracket, but we were unsuccessful in this task for numerous reasons. First, the design of the fixture caused a premature impact between the fixture and the bracket, thus testing the integrity of the fixture, not the restrictor. This was remedied by removing material from the fixture and changing the orientation of the bolt at the large end of the link. When the test was re-run, the same premature impact was observed, and the only solution to this issue was to remove the third attachment point to the transmission case, which involved removing both the bolt and some of the bracket material. The next test run was successful in the fact that the integrity of the component was tested, but due to the deflection of the rubber damper, the bracket did not fracture and withstood more than 13500 lbf. This is significantly higher than the 3400 lbf that the part is designed to withstand in compression. We attempted to raise the speed to 12 mph for the subsequent test run, but this resulted in the shearing of connection bolts and damage to the test fixture. From the first stages of testing, we determined that finding the fracture pattern of the current design would be impossible, so we began to make modifications to the bracket, becoming more and more aggressive with each subsequent run.

During the testing process, we conducted a second hoist review to re-familiarize ourselves with the undercarriage and drivetrain of the Edge. After examining the subsystem again, we determined to focus on causing a failure at the point where the link and the bracket attach, as this is the only single-point attachment in the system. Any other fracture location would require two or more bolt holes to fail, causing more of a design difficulty.

Photographs of the progression of the modifications can be seen in Appendix D. After removing much material from the bracket and cutting slots and/or holes in the plate attached to the link, we finally saw failure in the bracket, but unfortunately, it was located at a transmission attachment bolt hole, which is not the failure location we were pursuing. Additional variations of the material removal were tested, which resulted in loads between 8000 and 14000 lbf before bottoming out or failure in the same transmission attachment.

For the next run, the material removal was even more aggressive and the slots and holes in the plate were replaced with a keyway in the general direction we were hoping to force the link. Although the keyway did not work as planned, we observed failure in the bracket in the anticipated location at the link bolt. The same bracket material removal was repeated for the last five runs with variations to the plate. Complete failure in the plate was not achieved, but the load at bracket fracture ranged from 5100 lbf to 5900 lbf, which is above the design specification of 3400 lbf. The final test run led to the determination of our alpha design.

FINITE ELEMENT ANALYSIS

While component-level testing was being conducted, Morteza Tanbakuchi, a Ford engineer, was helping us to determine the forces that the roll restrictor experienced during an NCAP test. Using CAE modeling, the roll restrictor within the model will be isolated and the loads it endures from impact will be calculated. Although the model is not precise, this analysis will provide us with a general upper bound for our design. Using this force value and the design specifications, the redesign must stay within this force window under compression.

Once the general material removal and geometric alterations to the bracket and link are determined, we will implement those changes to the CAE model and run simulations and FEA to confirm our new design. We would also like to explore manufacturing the bracket from aluminum instead of cast iron.

ALPHA DESIGN

Based on the results of the testing and hoist reviews, we determined that the best alpha design incorporated aggressive material removal to force a fracture point on the bracket where the link attaches. At this bolt on the link, the plate was also weakened to try to encourage a complete separation of the bracket from the link, thus providing the maximum amount of free space once the restrictor breaks, which would interrupt the load path. Figure A8 on page A13 is the pre-test picture of our alpha design. The plate has had side cutouts surrounding the bolt hole that connects the link to the bracket, and a series of four holes have been drilled out to weaken the plate to cause a complete division between the two parts. The material removed from the bracket can be compared to the CAD rendering in Figure A9 on page A13. Most of the material removal on the bracket was concentrated at the connection to the link, which is circled on the photograph.

Proprietary Information

Figure A8. Pre-test photo of alpha design. Connection to link circled.

Proprietary Information

Figure A9. CAE rendering of original bracket.

Figure A10 and Figure A11, page A14, are post-test pictures of the alpha design. As seen circled in both photos, the bracket broke where planned (Figure A10), but the plate did not completely separate (Figure A11). The third transmission attachment point on the bracket had to be removed due to fixture limitations, but our customer requirements necessitate all three attachments, so our alpha design will incorporate this missing section.

Proprietary Information

Figure A10. Post-test left-side view of roll restrictor.

Proprietary Information

Figure A11. Post-test right-side view of roll restrictor.

Figure A12, page A15, shows the preferred motion the roll restrictor will experience during impact. The force of the crash will break the bracket and the plate surrounding the link bolt, causing a complete separation of the link from the bracket, which will in turn swing downward and out of the way, interrupting the load path.

Proprietary Information

Figure A12. Proposed motion of link after impact.

PROJECT PLAN

In order to allow enough time to prove out the engine roll restrictor design, the milestones for the project must be completed as quickly as possible. The project plan is summarized in the Gantt chart shown in Appendix E. These milestones include: component-level testing, FEA and CAE analysis, determining the final design, and prototype testing. In addition to these milestones, the team will simultaneously be working on required submissions and presentations for the class.

Literature analyses and benchmarking were conducted to determine what other types of engine roll-restrictors are on the market. In addition, meetings with engineers at Ford were held, and expectations as well as budget constraints were clearly defined. In a meeting with Mr. Kozak on January 16, 2008, he specified that he would like the prototype tested on two vehicles, if possible, by the end of the semester. This demonstrates the need for an accelerated time line for the group.

Component-level testing was conducted at the Ford Safety Innovations Labs. The testing was used to determine the loads that the part withstands during a full-vehicle impact. We modified several parts for testing to see which one would fracture in a predetermined manner while enduring the normal operating loads. The testing took over a week to complete due to problems with the fixture.

The results from the component-level testing were analyzed through engineering analysis and used to generate an alpha design concept. The alpha design is a preliminary concept that will be tested through a FEA and CAE analysis to determine whether the proposed part can withstand fatigue and stress. In addition, the analysis will provide the load history of the part by showing the load cycles the part undergoes during normal operation. As a team, we have to investigate durability standards set by Ford to ensure that our part meets them.

Following confirmation of our proposed design through FEA and CAE analysis, the engine roll restrictor must be signed off by Ford engineers in order to undergo prototyping. Once the part is fabricated, it will be implemented in two Ford Edges where it will endure a barrier test. The vehicles have been ordered and the necessary documentation must be filled to specify the instrumentation and cameras required. After the crash, we will analyze the results to see if the new part generates a decrease in G-force deceleration pulses.

The budget for this project is $100,000. This includes testing, parts, labor, two Ford Edges, drawings, and any other components needed to develop a suitable engine roll restrictor. The team has stayed within budget and does not see a need to surpass it.

INFORMATION SOURCES

The following loads for the engine roll restrictor and link were determined through meeting with Paul Jabbour, a Design and Release engineer (D&R), at Ford Motor Company: load in tension, 6800 lbf, load in compression, 3400 lbf, and resonant frequency, 500 Hz. The roll restrictor and the link must also include the same fastening locations on the transmission case and one fastening location to the sub-frame, respectively.

A hoist review and a cross-vehicle comparison were performed with Steve Brumley at the Ford Livernois facility. The hoist review allowed for the visualization of an engine roll restrictor in a complete vehicle, as well as allowing the team to become familiar with other key components on the vehicle such as the rear driveshaft. While at the Livernois facility, a comparison between the engine roll restrictor on the Ford Edge and on a Ford Escape and Ford Fusion was conducted. A view of the roll restrictor on the Ford Edge is seen in Figure A13, page A17. Due to confidentiality, pictures of the Fusion and Escape cannot be shown.

Proprietary Information

Figure A13. View of Engine Roll Restrictor on Ford Edge

A study by Wellman et al. [1] discussed the implementation of E-glass fabric, a type of fiberglass, for an engine roll restrictor. A Finite Element Analysis (FEA) was used to evaluate changing a restrictor from steel to E-glass fabric. It was determined that the E-glass fibers would not be sufficient to meet the torsional requirements for the same design as steel but with E-glass fibers. Therefore, a web across the center was added to increase torsional stiffness. Through implementing automated design optimization software, the orientation of the E-glass fibers and a final shape was determined as shown in Figure A14 below. The ribs along the upper surface were used to further strengthen the component as determined by the FEA.

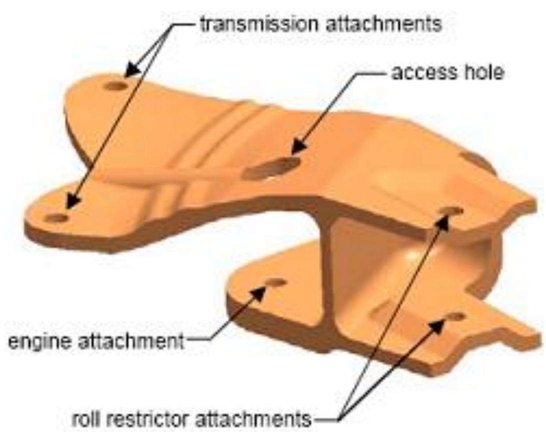

Figure A14. Composite engine roll restrictor

U.S. Patent No. 6,729,430 to Adams et al. [2] discloses an engine roll restrictor that prevents motion of a transversely mounted engine. The engine roll restrictor described in this patent is only able to carry a load in tension. The load is in tension when the vehicle in operating in the forward direction and the roll restrictor is experiencing a compressive load when the vehicle is operating in the reverse direction. This means that the engine is able to move more freely when the vehicle is in reverse, when the load would be in compression. Figure A15 and Figure A16, page A16, show a representation of an embodiment of this invention.

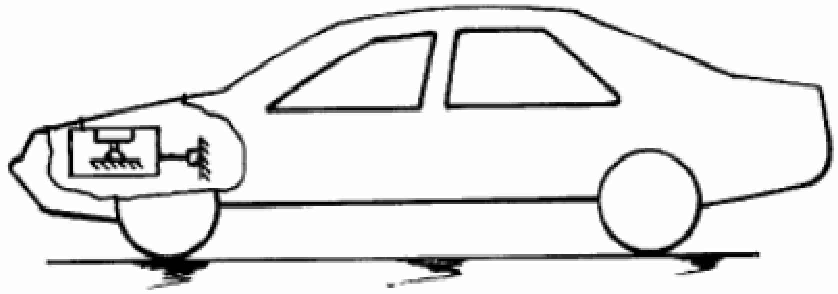

Figure A15. Sample car from '430 patent

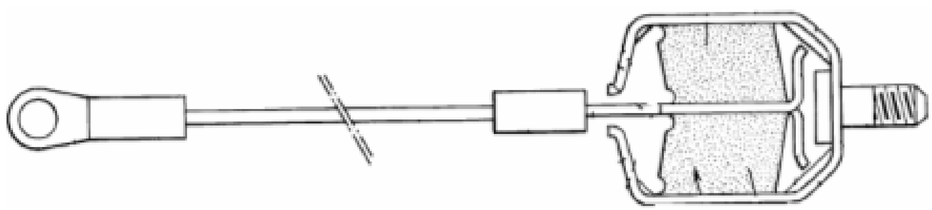

Figure A16. Representative drawing of engine roll restrictor from '430 patent

U.S. Patent No. 6,880,667 to Gotou [3] discloses an engine mount that allows energy to be absorbed by a side member during impact. The engine mount is located on one of the front side members on a frame of a vehicle as shown in Figure A17. In prior art, the support member could not collapse properly on impact and dissipate energy due to a mount being attached to the side member. Therefore, the mount as shown in Figure A18, page A19, is capable to detach from the side member during impact. The mount has special failure locations that allow the mount to detach at certain loads.

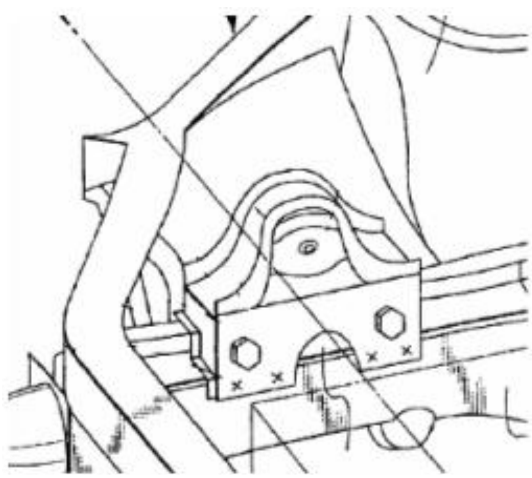

Figure A17. Representative mount from '667 patent.

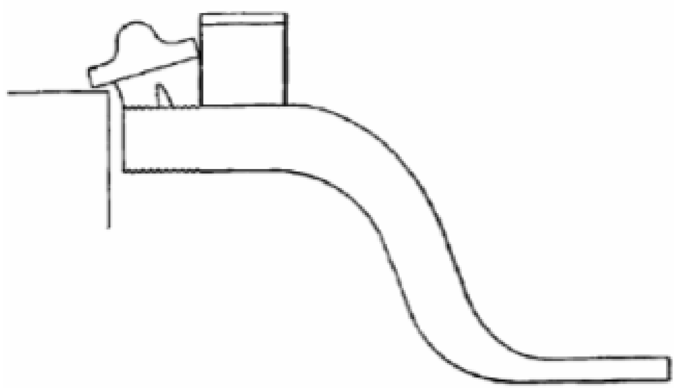

Figure A18. Failure method of '667 engine mount.

U.S. Patent No. 4,901,814 to van Broock et al. [4] discloses a way to prevent engine roll in a transversely mounted engine. The invention includes a pair of arm members 18 and 19 that attach to an upper and lower portion of an engine and to an upper and lower portion on a vehicle frame. These two attachment means prevent the engine from large displacements during large torque outputs by the engine. Figure A19 below, is a representation of an embodiment of the invention.

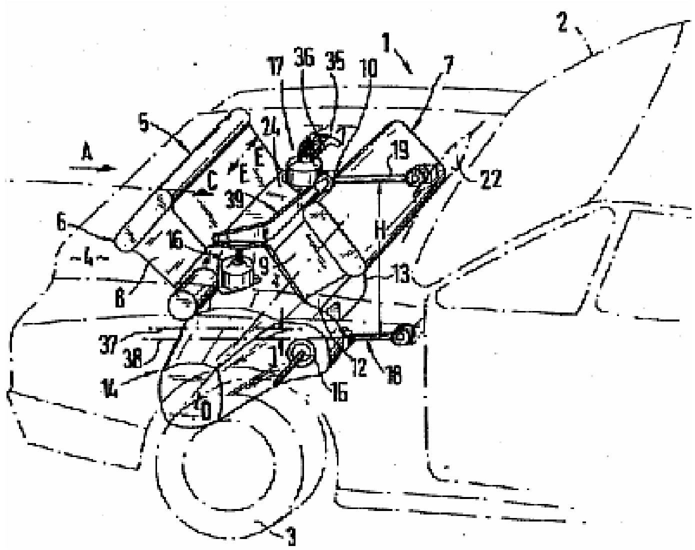

Figure A19. Representative drawing for vehicle with '814 engine roll restrictors

ENGINEERING ANALYSIS

Several engineering specifications will be considered during the design process of the engine roll restrictor. Our initial evaluation will begin with fracture mechanics. The geometry and material properties play a crucial roll in the failure modes and patterns on the bracket. Furthermore, weakening strategies can be determined through fracture mechanics that will allow the part to fracture during crashes and prevent loads from being transferred to other parts of the vehicle.

To evaluate the failure modes of the part, component-level testing was performed on the original bracket and link as well as concepts designs. The testing provided estimated loads at failure for the various design concepts. Video analysis of recordings taken during the testing allows the fracture method to be analyzed as well as allow for the observation of the deflection experienced by the bracket and link during impact.

A dynamic analysis of the link and bracket will play a decisive roll in our design goals. Through a finite element analysis, the internal dynamics of the parts will be captured, which will provide further insight into the functionality of the part before it is tested on a full vehicle. This analysis will also give a more realistic representation of the behavior of the part compared with a simplified analysis using Newton's Second law.

Engineering models will provide us with further specifications than those initially provided to us by our sponsor. In particular, the CAE simulation performed on a full vehicle will provide an estimate of the load the link and bracket will experience during a crash. This estimate will provide a range of loads experienced by the link and bracket that will allow for a more accurate design.

To provide proof of concept, a working prototype of the alpha design will be manufactured and installed on a full vehicle for a full frontal NCAP test. This is the same test that produced high deceleration pulses on previous models. By reducing the deceleration pulses to an acceptable range (still to be determined), and showing its durability, our concept will be proven.

The design drivers behind our design are the link, plate, and the bracket. These parts have been determined to be crucial to achieving our design goal of reducing the deceleration pulses experienced during a full frontal impact test.

Several aspects of our design are providing challenges. In particular, designing a part that will fail during impact and still be able to meet the durability requirements could prove to be a challenge. The major problems expected during the completion of our project are:

- Reaching project milestones on time
- Customers' changing needs
- Coordinating projects with Ford engineers

All of these problems will be addressed by having weekly meetings with our design sponsor. Through these meetings, we will be able to schedule the appropriate meetings with relevant Ford departments in order to meet our mile markers to complete the project.

CONCLUSIONS

Based on our component-level testing, we have determined an alpha design for the roll restrictor that involves aggressive material removal on both the cast iron bracket and the link plate (Figure A8, page A13). Engineering specifications have provided us with a lower bound for force, and we are currently awaiting the results from a CAE model of an NCAP test to determine an upper bound for our redesign. We are planning on incorporating our design changes into the current CAD models so that FEA can be conducted before the design is finalized. We will also be investigating durability issues with such aggressive changes to the bracket.

REFERENCES

[1] Wellman, Scott, Johanna Burgueno, and Ron Averill. 1-5. 18 Jan. 2008, "Advanced technologies for design and fabrication of composite automotive components," <http://www.redcedartech.com/ss/papers/CompositeComponents.pdf>.

[2] US6,729,430 (2004-5-4) Adams et al, Engine motion restrictor for vehicle, and vehicle including such an engine motion restrictor.

[3] US6,880,667 (2005-4-19) Hiromitsu Gotou, Impact absorbing structure for vehicle body and related method.

[4] US4,901,814 (1990-2-20) van Brook et al., Mounting of an internal combustion engine.

APPENDIX B. FUNCTIONAL DECOMPOSITION

Level 1:

1. Transmit motion from driveshaft to rear axles
2. Connect to subframe at four different points
3. Move in longitudinal direction during impact
4. Allow wheels to rotate at different speeds

Level 2:

1. Transmit motion from driveshaft to rear axles
    - 1.1 Take rotational motion as input in one direction
    - 1.2 Output rotational motion in two directions, opposite to each other, and orthogonal to the input motion
2. Connect to subframe at four points
    - 2.1 Two connection locations at rear of RDU on either side of unit
    - 2.2 Two connections on front side of unit
    - 2.3 Connections do not impede unit motion during impact
3. Move in longitudinal direction during impact
    - 3.1 Front connections release unit upon impact
    - 3.2 Rear connections do not cause interference between unit and subframe
    - 3.3 Minimum interference between unit and subframe
4. Allow wheels to rotate at different speeds
    - 4.1 Take wheel resistance as input
    - 4.2 Take driveshaft motion as input
    - 4.3 Output motion to rear axles based on wheel resistance

APPENDIX C. NEW DESIGN CONCEPTS

- Weaken the brackets supporting the RDU
  - Add notches in the front casting to make fracture more likely during impact
  - Add a perforation along the line where the front casting naturally fails so that it will fail at a smaller load
  - Add a keyway where RDU attaches to rear brackets that will separate on impact
  - Add notches to reinforcement of rear brackets so that they fold in half on impact
- Weaken the bolts attaching the RDU to brackets
  - Remove one of the two bolts connecting the rear mounting bracket to the RDU so the force of impact would be only on one bolt instead of two, making it more likely to shear.
  - Use weaker bolts to attach RDU at all attachment points
- Reduce distance needed for driveshaft crash disks to break out keyways
  - Make distance needed for crash disks to break out keyways smaller so less clearance is needed between RDU and subframe
  - Make distance between crash disks and keyways zero so that clearance needed between RDU and subframe little to none
- Redesign subframe
  - Remove material from subframe where interference with RDU could possibly occur
  - Change subframe so that deformation of subframe during impact would prevent interference with RDU
- Give RDU more longitudinal freedom
  - Make rear mounting brackets shorter and use longer bolts to attach the brackets to subframe so that the brackets do not interfere with the subframe during impact
  - Replace rear mounting brackets with a rod, or something similar, that is rigidly attached to the RDU, and runs through the bushings on the subframe, giving the RDU vertical support, and allowing longitudinal freedom
- Use a different material for one or more of the parts
  - Make front casting out of different material that is more likely to fail at the forces seen during impact
  - Make rear mounting brackets out of different material that is more likely to fail at the forces seen during impact
- Active mechanism to destroy/move part of the RDU mount
  - Create a pin that falls out on impact to connect the RDU to rear mounting brackets
  - Use bolts that explode on impact so entire assembly falls apart, preventing RDU from interfering with subframe before crash disks break away keyways

APPENDIX D. PUGH CHART

Design Criteria	Weight	Original	Weaken the brackets attaching the RDU	Weaken the bolts attaching the RDU	Reduce distance needed for driveshaft crash disks to break out keyways	Redesign subframe	Give RDU more longitudinal freedom
Manufacturing Cost	1	0	0	+	0	0	+
Implementation	3	0	+	+	- -	- - -	0
Durability	2	0	-	-	- -	-	- -
Prevents RDU from Impacting the Subframe	3	0	+	+	0	-	0
Manufacturability	2	0	0	0	0	0	+
Weight	2	0	+	0	0	+	0
	+	0	8	7	0	2	3
	0	11	2	4	0	3	8
	-	0	2	2	10	14	-4
	Total	**0**	**6**	**5**	**-10**	**-12**	**-1**

APPENDIX E. ENGINEERING DRAWINGS OF FINAL DESIGN

Proprietary Information

Proprietary Information

APPENDIX F. GANTT CHART

Task / Week	1/6	1/13	1/20	1/27	2/3	2/10	2/17	2/24	3/2	3/9	3/16	3/23	3/30	4/6	4/13
Preparation for Design Review #1															
Preparation for Design Review #2															
Preparation for Design Review #3															
Preparation for Design Review #4															
Preparation for Final Report															
Weekly meetings with Steve Kozak															
Literature search and benchmarking															
Problem Analysis															
Preliminary meeting with engineers at Ford • Determine design specifications • Determine what deliverables are expected															
Functional Decomposition															
Brainstorming/Concept Generation															
Component Level Testing • Determine baseline loads • Replicate full vehicle event • Modify parts for testing • Determine loads for modified parts • Create repeatable fracture pattern on part • Video/photo/data collection															
Engineering Analysis • Review of component level data															
Alpha Design Concept • Integrate physical modifications to existing CAE model • Run FEA to determine if new design meets design specifications • Determine if new design withstands durability requirements															
New Direction of Project-Focus on Rear Differential Unit															
CAE Analysis • Dimensional/Clearance analysis • Pulse matching • Simulations of modifications • Stress concentration analysis															
Brainstorming/Concept Generation															
Vehicle Preparation • Write test planning form • Inspect vehicle • Order replacement parts • Determine instrumentation and cameras needed • Make modifications to upgrade vehicle to Lincoln MKX • Make modifications to RDU brackets and rear subframe • Conduct walk around at Livernois • Enter barrier queue															
Crash Test															
Post Test Analysis															
Design Expo															

APPENDIX G. BILL OF MATERIALS

Item	Quantity	Supplier	Part #	Cost	Contact	Notes
BRKT-RR AX DIFF MTNG TO C/MBR	1	PROGRESSIVE STA	7T43-4K204-AC	$2.57	J. FAZIO	Left-hand rear RDU mounting bracket
BRKT-RR AX DIFF MTNG TO C/MBR	1	PROGRESSIVE STA	7T43-4K204-BC	$3.40	J. FAZIO	Right-hand rear RDU mounting bracket
BRKT RR AX DIFF	1	PROGRESSIVE STA	7T43-4K360-AE	$6.23	J. FAZIO	Right-hand front RDU mounting bracket
BRKT RR AX DIFF LH	1	PROGRESSIVE STA	7T43-4C037-AE	$6.23	J. FAZIO	Left-hand front RDU mounting bracket

APPENDIX H. DESIGN CHANGES SINCE DESIGN REVIEW #3

One major change was made to each bracket since Design Review #3. The alpha design in DR3 involved shortening each of the original two brackets and using longer bolts, supported by a sleeve, to attach them to the subframe. This idea can be seen in Figures H1 and H2. The elongated bolt would have extended from where the material had been removed, been supported by a sleeve, and attached by a bolt to the subframe in the same manner as the original design.

Proprietary Information

Figure H1. Material removed from left bracket for alpha design.

Proprietary Information

Figure H2. Material removed from right bracket for alpha design.

This purpose of this design was to allow the bolts to push out the bushings in the subframe during an impact, therefore allowing the RDU to have freedom of motion. It was feared, however, than if the bolts and their respective sleeves were to impact the subframe bushing at an angle slightly off of horizontal that they would become jammed in the bushing. This would prevent free movement of the RDU, and defeats the purpose of the redesign.

The final design replaces the bolt with a rod that is rigidly attached to the shortened bracket. This eliminates the need for the sleeve for additional support, eliminating some of the chance that the assembly becomes jammed in the bushing during impact. Finally, the rod is made to extend through the bushing and left free. This allows the bracket to support the RDU while giving it freedom in the longitudinal direction. Finally, since the rod is already completely through the bushing, unless the RDU moves in such a way to put the rod at a very extreme angle, it will not jam into the bushing during impact. Views of the final design similar to Figures H1 and H2 can be seen in Figures H3 and H4 for comparison.

Proprietary Information

Figure H3. Final design of left bracket for comparison to Figure H1.

Proprietary Information

Figure H4. Final design of right bracket for comparison to Figure H2.

APPENDIX I1. MATERIAL SELECTION ASSIGNMENT

The material to be used to manufacture the redesigned rear brackets was selected using the Cambridge Engineering Selector (CES) software. In doing this, we began by identifying the function, objective, and constraints of the redesigned RDU brackets, all of which are listed below.

Function:	Support the RDU
Objective:	Prevent rotation of RDU relative to subframe while maintaining five point connection to subframe
Constraints:	Bracket must maintain original connection points to RDU, fit in space provided, and not undergo any permanent deformation during operation

The material indices were selected based on the specifications outlined for the current part, and their respective limits were selected based on the material from which the brackets are currently manufactured. The final material selected must meet the property limits listed below.

Yield Strength:	44-58 ksi
Tensile Strength:	58-67 ksi
Maximum Elongation:	22-28%

From this, the choice for material was narrowed to twenty choices, all of which satisfy the property limits. The top five of these were chosen based on typical uses of each material as listed in the CES software and are shown in Figure I1, page I2. These five choices are listed below.

- Low alloy steel: AISI 4620
- Carbon steel: AISI 1022
- Carbon steel: AISI 1030
- Carbon steel: AISI 1118
- Wrought aluminum alloy: 2026-T3511

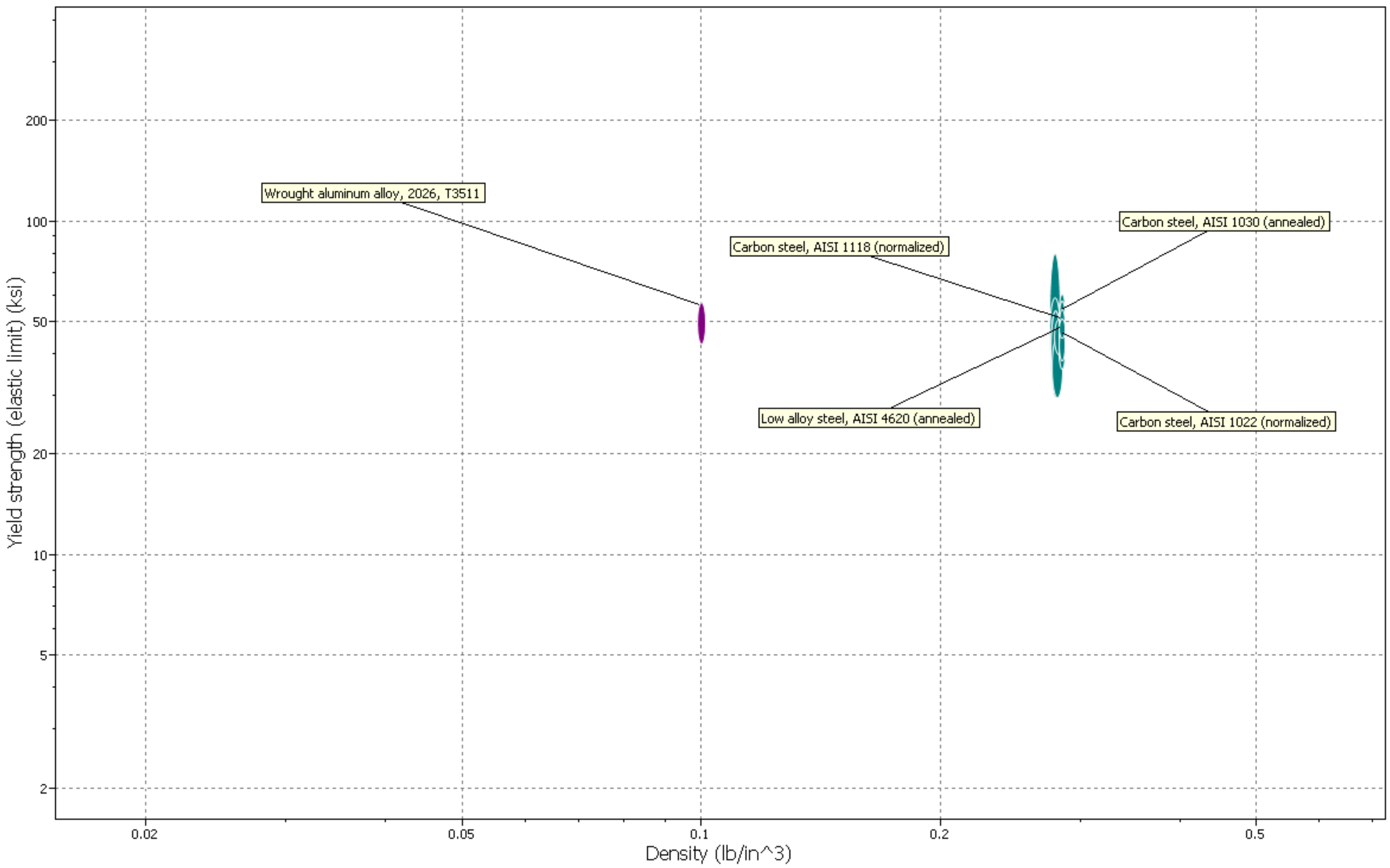

Figure I11. Yield strength vs. density for materials that satisfy property limits. Top five choices are called out.

The final choice to manufacturing the redesigned brackets is a low alloy steel, AISI 4620. This steel is very similar to the parts original material. It falls within all of the property limits defined above with a yield strength of about between 48 and 60 ksi, ultimate tensile strength of between 66 and 81 ksi, and a maximum elongation of between 24 and 38%.

APPENDIX I2. DESIGN FOR ASSEMBLY

Using Boothroyd and Dewhurst (B-D) Design for Assembly (DFA) charts, we determined the assembly efficiency of our original and redesigned bracket assemblies. The assembly efficiencies for the original and redesigned rear brackets are summarized in Table I1 below.

To test for theoretical minimum number of parts, we determined whether parts moved relative to one another, whether parts needed to be made of different materials, whether combinations of parts prevented assembly or disassembly of other parts, and if servicing of the assembly would be adversely affected by integration. If none of these criteria were necessary, a part could be integrated or eliminated.

After analyzing the design, the team determined that by integrating the bracket and rod of our assembly, we can increase the efficiency by eliminating the welding operation necessary to fasten the rod to the bracket. To integrate the two assemblies, we propose to die-cast the part into one piece. The DFA charts are provided on pages I4 and I5 for the original bracket assembly and the redesigned assembly.

	Original Design Assembly Efficiency	**Redesign Assembly Efficiency**
Rear Left Bracket	0.188	0.257
Rear Right Bracket	0.235	0.321

Table I1. Assembly Efficiencies for Original Design and Redesigned Brackets

Design for Assembly Original Design B-D Chart

Rear Left Bracket

1	2	3	4	5	6	7	8	9	Name of Assembly
Part ID No.	number of times the operation is carried out consecutively	two-digit manual handling code	manual handling time per part	two-digit manual insertion code	manual insertion time per part	operation time, seconds [2] * [[4] + [6]]	operation cost, cents 0.4 * [7]	figures for estimation of theoretical minimum parts	
1	1	30	1.95	00	1.5	6.9	2.76	1	Bracket
2	1	10	1.5	96	12	27	10.8	0	Rod
3	2	10	1.5	38	6	15	6	2	Bolts
4	1	01	1.43	08	6	14.86	5.94	1	Nut
					Total	63.76	25.5	4	Assembly Efficiency=3*NM =
						TM	CM	NM	TM 0.188

Rear Right Bracket

1	2	3	4	5	6	7	8	9	Name of Assembly
Part ID No.	number of times the operation is carried out consecutively	two-digit manual handling code	manual handling time per part	two-digit manual insertion code	manual insertion time per part	operation time, seconds [2] * [[4] + [6]]	operation cost, cents 0.4 * [7]	figures for estimation of theoretical minimum parts	
1	1	30	1.95	00	1.5	6.9	2.76	1	Bracket
2	1	10	1.5	96	12	27	10.8	0	Rod
3	3	10	1.5	38	6	15	6	3	Bolts
4	1	01	1.43	08	6	14.86	5.94	1	Nut
					Total	63.76	25.5	5	Assembly Efficiency=3*NM =
						TM	CM	NM	TM 0.235

Design for Assembly Redesign B-D Chart

Rear Left Bracket

1	2	3	4	5	6	7	8	9	Name of Assembly	
Part ID No.	number of times the operation is carried out consecutively	two-digit manual handling code	manual handling time per part	two-digit manual insertion code	manual insertion time per part	operation time, seconds [2] * [[4] + [6]]	operation cost, cents 0.4 * [7]	figures for estimation of theoretical minimum parts		
1	1	30	1.95	21	6.5	16.9	6.76	1	Bracket/Rod	
2	2	10	1.5	38	6	15	6	2	Bolts	
3	1	01	1.43	08	6	14.86	5.94	1	Nut	
					Total	46.76	18.7	4	Assembly Efficiency=3*NM =	
						TM	CM	NM	TM	0.257

Rear Right Bracket

1	2	3	4	5	6	7	8	9	Name of Assembly	
Part ID No.	number of times the operation is carried out consecutively	two-digit manual handling code	manual handling time per part	two-digit manual insertion code	manual insertion time per part	operation time, seconds [2] * [[4] + [6]]	operation cost, cents 0.4 * [7]	figures for estimation of theoretical minimum parts		
1	1	30	1.95	21	6.5	16.9	6.76	1	Bracket/Rod	
2	3	10	1.5	38	6	15	6	3	Bolts	
3	1	01	1.43	08	6	14.86	5.94	1	Nut	
					Total	46.76	18.7	5	Assembly Efficiency=3*NM =	
						TM	CM	NM	TM	0.321

APPENDIX I3. DESIGN FOR ENVIRONMENTAL SUSTAINABILITY

Using the CES software, we determined five best materials to be used on our brackets based on the function, objectives, and constraints of our part. We then performed an assessment on the top two material choices to see how they would impact the environment. Using SimaPro, we compared a low alloy steel, AISI 4620 and wrought aluminum alloy, 2026-T3511. Since these materials were not available in the software, we chose the closest materials available: Steel low alloy ETH S and AlCuMg2 (2024) I.

In Figure I2, page I8, a comparison of total mass emissions between the two materials is shown. From this plot, the wrought aluminum alloy has a greater total number of emissions in all fields: raw materials, air, water, and solid waste.

Figure I3, page I9, shows the emissions in terms of disability adjusted life years (human health), potentially disappeared species fraction (eco-toxicity), and megajoules surplus (resources). Table I2, below shows the material choice that has a bigger impact within each of the EcoIndicator 99 damage classifications. The shaded boxes indicate the material with the greatest impact.

	Minerals	Land Use	Acidification/ Eutrophication	Ecotoxicity	Ozone layer	Radiation	Climate change	Resp. inorganics	Resp. organics	Carcinogens
Low Alloy Steel										
Wrought Aluminum Alloy										

Table I2. Material Choice with Greater Impact on EcoIndicator 99 Damage Classifications

The impacts are collapsed into the three meta-categories, human health, ecosystem quality, and resources. They are normalized with the average damage caused by an "average European person" over 1 year in Figure I4, page I10. The most important meta-category since it causes the most damage is resources. In this plot, wrought aluminum alloy causes the most damage. Finally, Figure I5, page I11, compares the material performances in Total EI99 Points. From this plot, we see that the low alloy steel is better than the wrought aluminum alloy.

From this analysis, we determined that the wrought aluminum alloy has the greater environmental impact. Considering the full life cycle of the product, both materials create a negative impact, however wrought aluminum alloy has a greater negative impact.

We took this analysis to the next step by considering an alternative material, carbon steel, AISI 1022 and compared it to the low alloy steel, AISI 4620. Since the exact carbon steel was unavailable in SimaPro and because the densities of all three carbon steel proposed by CES are similar, we extended this analysis to all three carbon steels: AISI 1022, AISI 1030, and AISI 1118. After performing an analysis using SimaPro, the carbon steel proved favorable over the low alloy steel. Figure I6, Figure I7, and Figure I8, page 12I12-I14, show the SimaPro analysis. Thus, we will present this new material to Ford for investigation in terms of mechanical properties as it shows to be better for the environment.

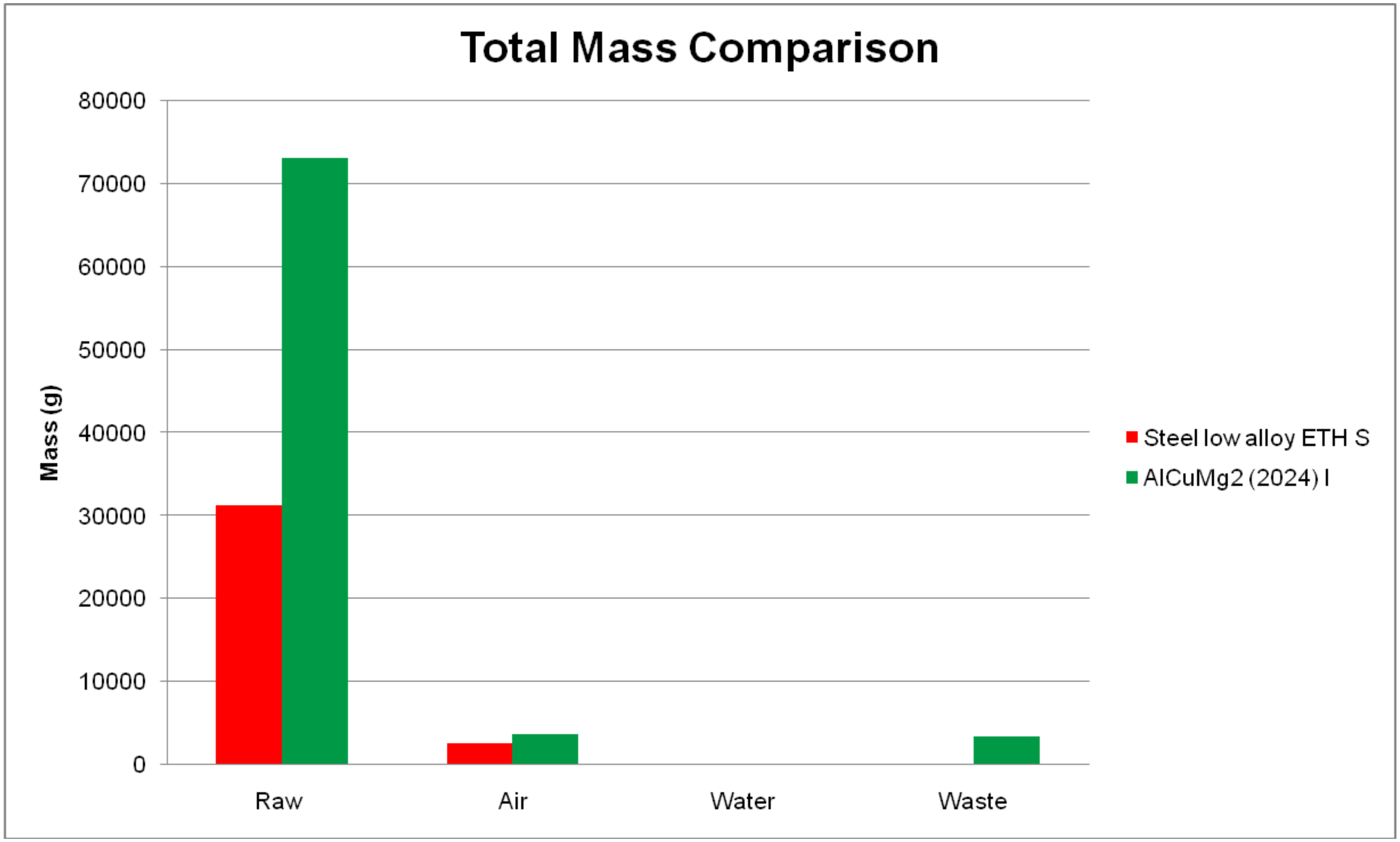

Figure I22. Total Mass Emission Comparison of low alloy steel and wrought aluminum alloy

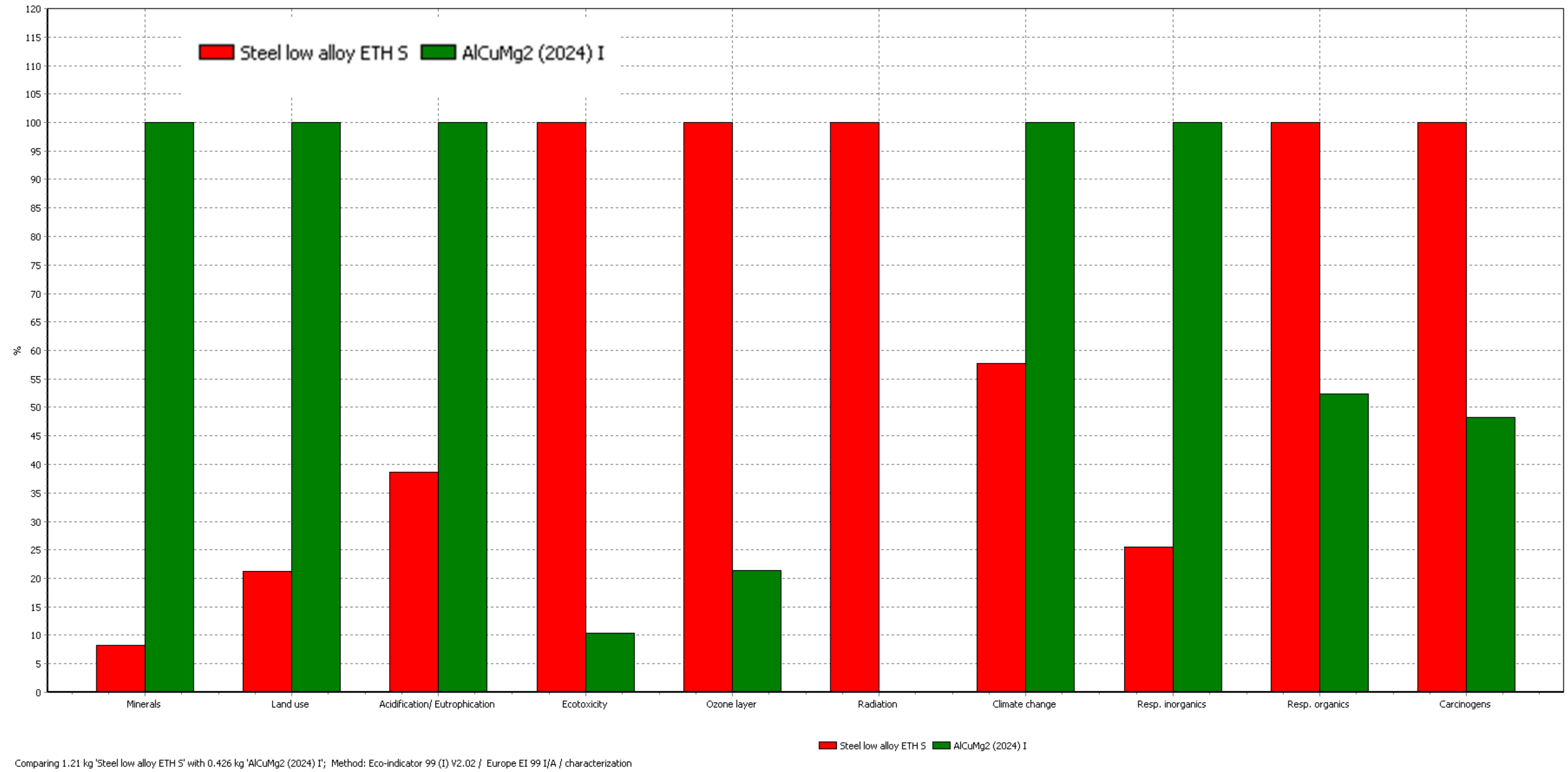

Figure I33. Comparison of low alloy steel and wrought aluminum alloy in EI99 Impact Categories

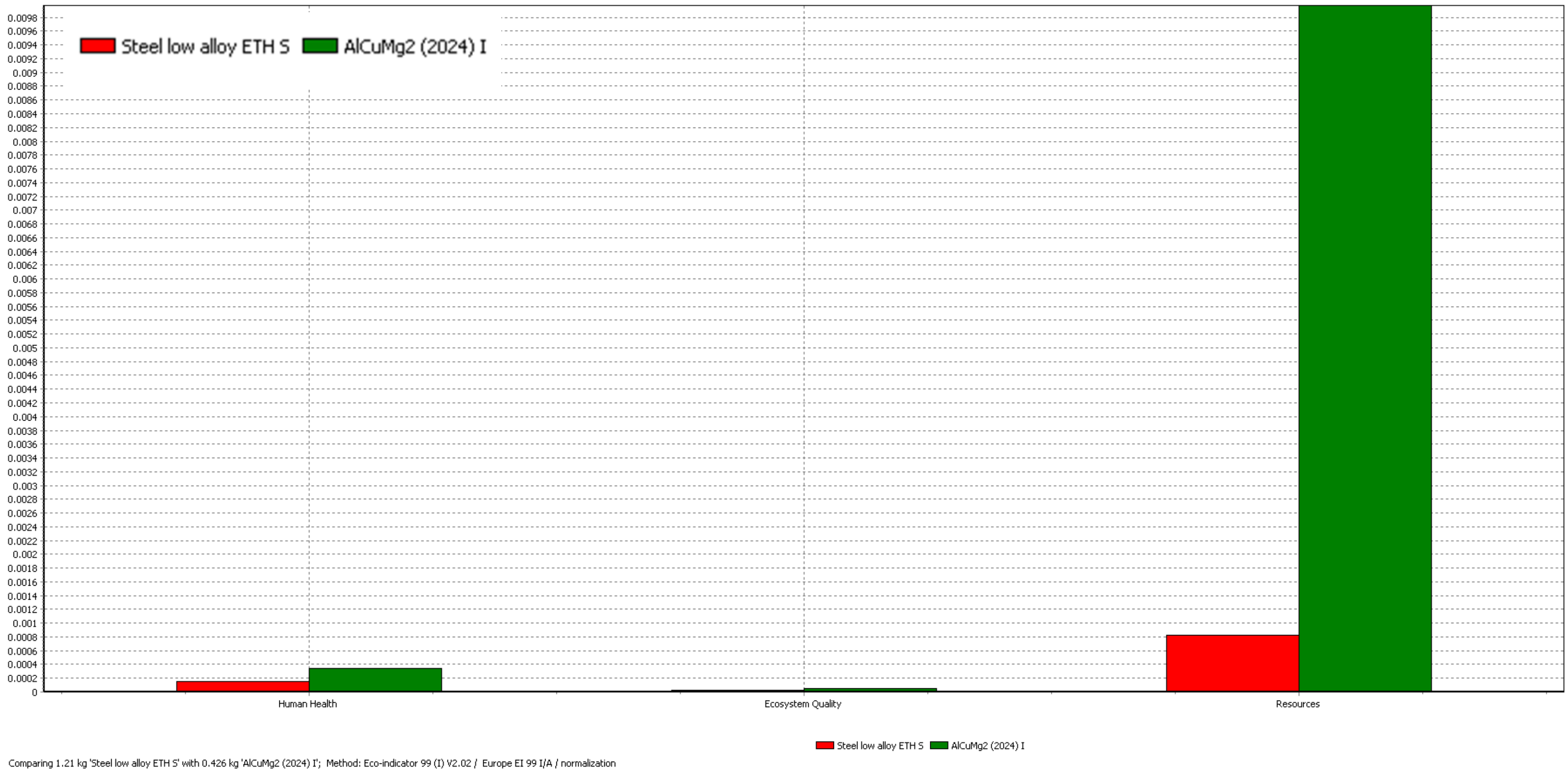

Figure I4. Comparison of low alloy steel and wrought aluminum alloy in Normalized "Meta-Categories"

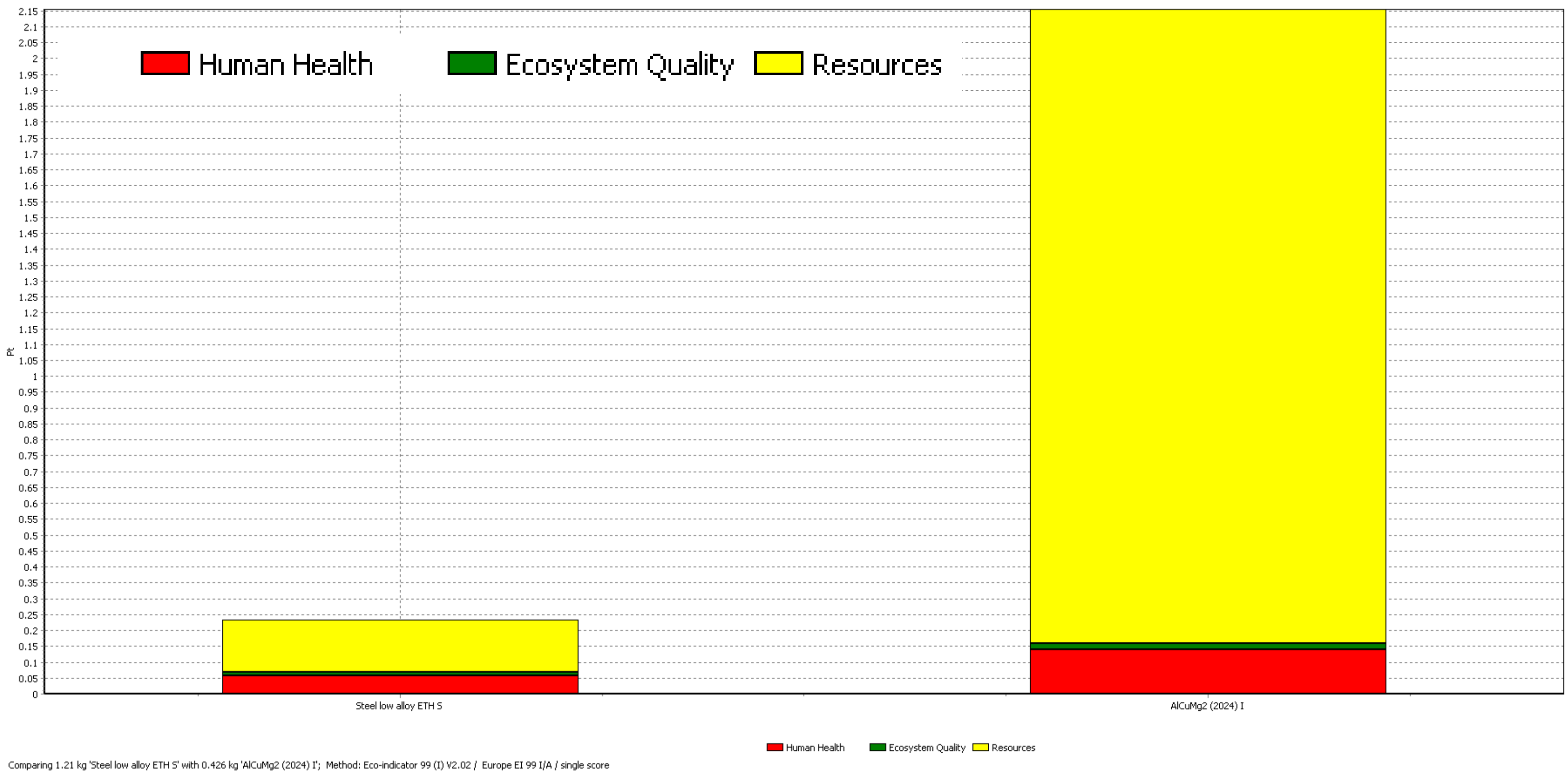

Figure I5. Comparison of low alloy steel and wrought aluminum alloy in Total EI99 Points

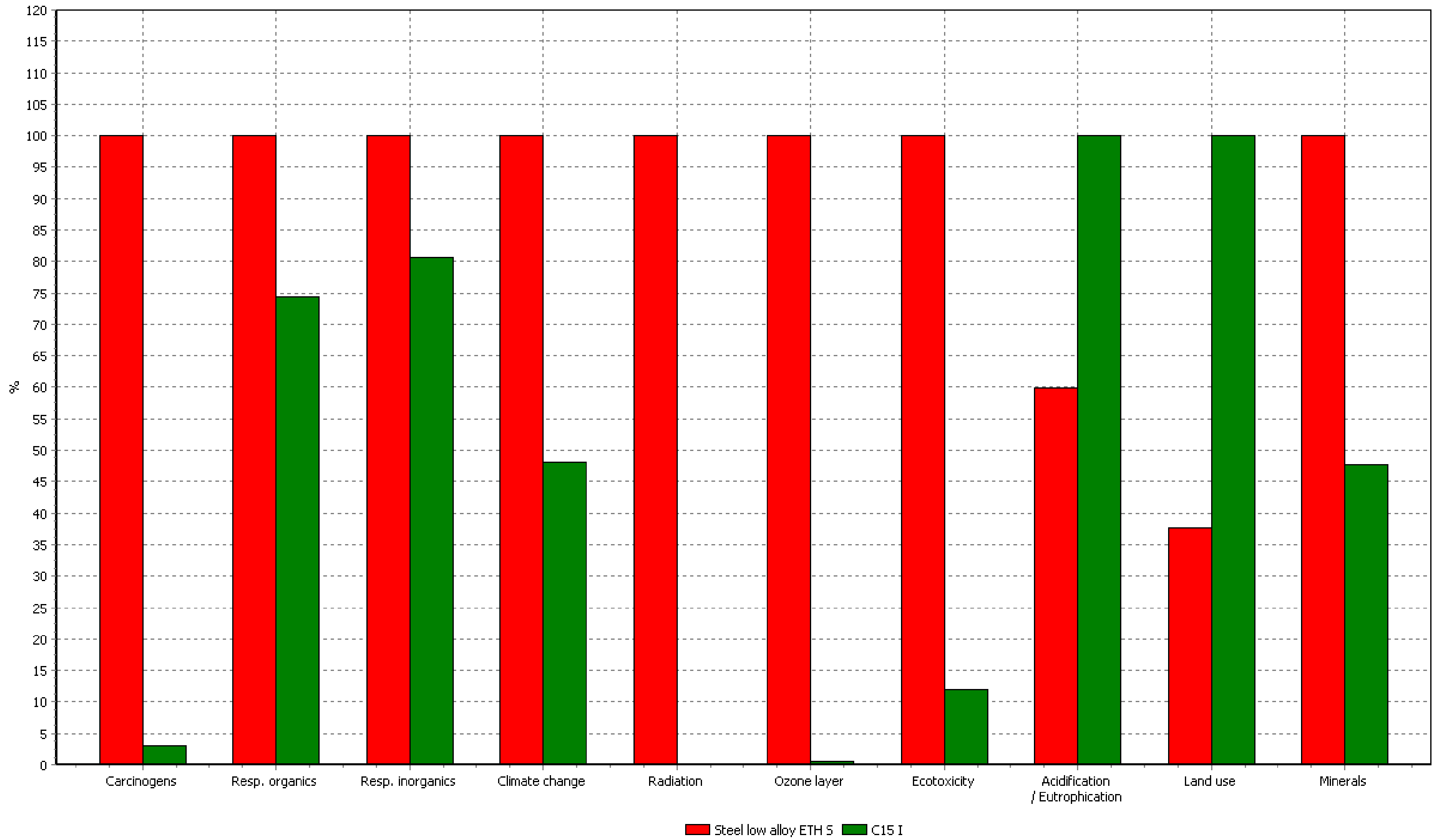

Figure I6. Comparison of low alloy steel and carbon steel in EI99 Impact Categories

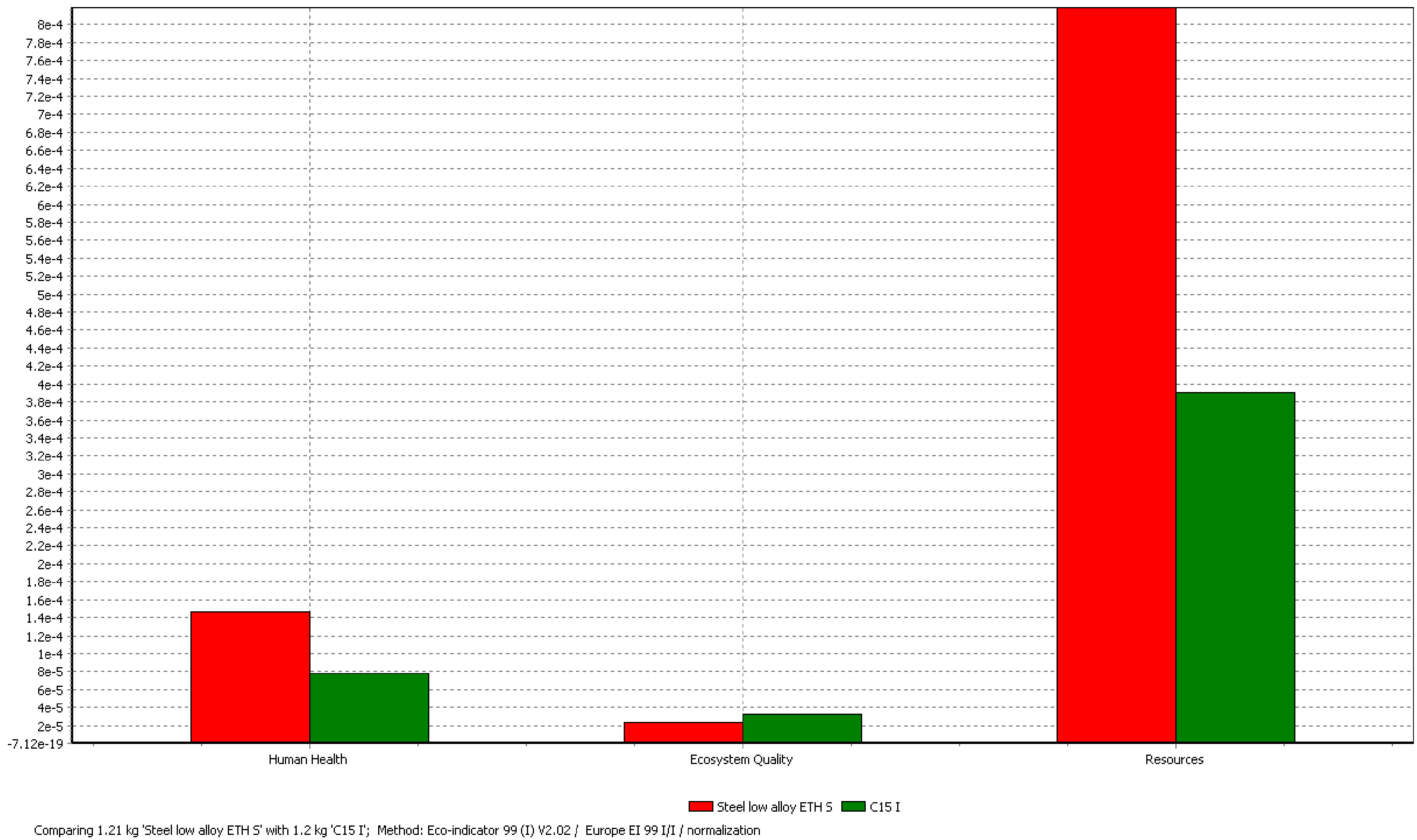

Figure I74. Comparison of low alloy steel and carbon steel in Normalized "Meta-Categories"

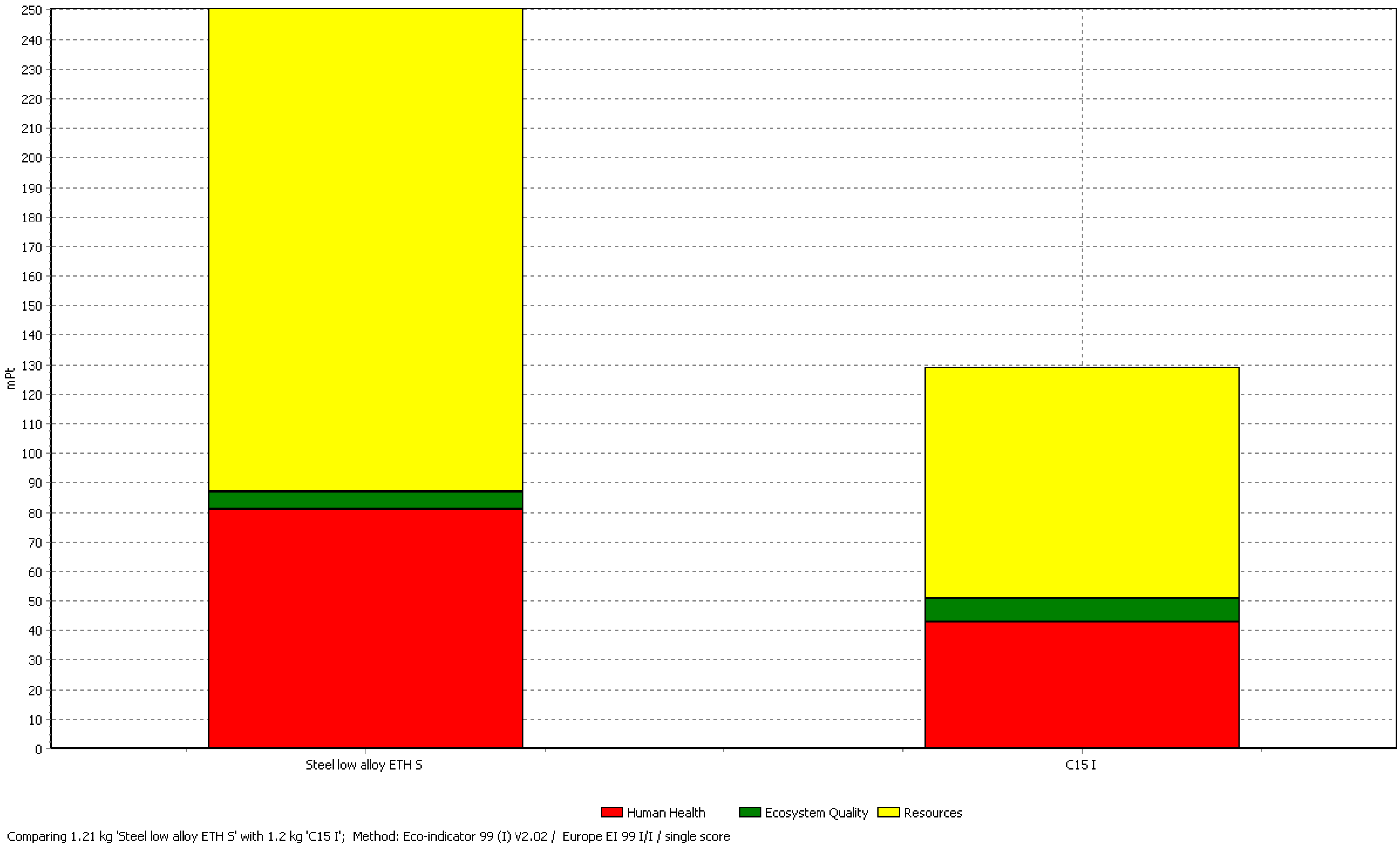

Figure I8. Comparison of low alloy steel and carbon steel in Total EI99 Points

APPENDIX I4. DESIGN FOR SAFETY

There are two assessments that can be conducted to evaluate how safe our part is: FMEA and risk assessment. The difference between FMEA and risk assessment is that FMEA is related to failure of components whereas risk assessment evaluates failure due to human error. A detailed risk assessment was conducted for our assembly to determine potential hazards installers might face during installation. A copy of the DesignSafe report can be found in Figure I9, page I16.

Most of the risks encountered during installation were mechanical while some were related to ergonomics and human factors.

The most serious mechanical risk encountered is if the vehicle falls of the lift during installation. While the likelihood of this event is unlikely, the risk level is moderate. The team developed a risk reduction method to reduce the risk level. By creating fixed enclosures/barriers for the vehicle, the risk level decreased to low, which is acceptable.

Another moderate mechanical risk is if an installer’s fingers get caught in the bracket during installation. This may be occasional due to the repetitive nature of installation, but by creating special fixtures or tools that guard the installer, this risk can be reduced to low.

The third moderate mechanical risk is if an installer stands up quickly and bumps their head on the undercarriage of the vehicle. We have suggested that installers wear head protection to reduce this risk to a low level.

Ergonomic and human factors play a role for two injuries that may occur: if the installer installs the part repetitively or if the part is installed improperly. Fatigue of the installer poses a moderate risk since repetitive installation for several hours may be detrimental to their health. To reduce this risk from moderate to low, we have suggested that job rotations be implemented. To prevent improper installation of the part, a risk which is low, standard procedures should be put into practice. This method will still keep the risk level at low.

After analyzing the DesignSafe results, we found no unexpected risks and all risks could be dealt with simple preventive measures.

While there is no such thing as a zero risk, in an ideal environment, the preventive measures placed would ensure that there would be no risk to the installer. The difference between an acceptable risk and zero risk with respect to function and safety is that an acceptable risk is one that is as low as reasonable practicable whereas a zero risk is non-existent. For our part, there were no zero risk hazards. However, our risk reduction methods reduced the risk levels to low.

designsafe Report

Application:	Crash Test	Analyst Name(s):	Team 9
Description:		Company:	ME 450
Product Identifier:		Facility Location:	
Assessment Type:	Detailed		
Limits:			
Sources:	Ford		

Guide sentence: When doing [task], the [user] could be injured by the [hazard] due to the [failure mode].

User / Task	Hazard / Failure Mode	Initial Assessment Severity Exposure Probability	Risk Level	Risk Reduction Methods /Comments	Final Assessment Severity Exposure Probability	Risk Level	Status / Responsible /Reference
All Users All Tasks	mechanical : crushing Vehicle falls off lift	Serious Remote Unlikely	Moderate	fixed enclosures / barriers	Slight Remote Unlikely	Low	
All Users All Tasks	mechanical : pinch point Fingers caught in bracket	Slight Occasional Possible	Moderate	special tools or fixtures	Slight Remote Unlikely	Low	
All Users All Tasks	mechanical : head bump on overhead objects Person stands up quickly during installation	Slight Occasional Possible	Moderate	head protection	Minimal Remote Unlikely	Low	
All Users All Tasks	ergonomics / human factors : lifting / bending / twisting Repetitive installation	Slight Occasional Possible	Moderate	job rotation	Minimal Remote Unlikely	Low	
All Users All Tasks	ergonomics / human factors : human errors / behaviors Improper installation	Slight Remote Unlikely	Low	standard procedures	Minimal Remote Unlikely	Low	

Figure I95. DesignSafe Report

APPENDIX I5. MANUFACTURING PROCESS SELECTION

The production volume of our brackets is based on the production of Ford Edges and Lincoln MKXs per year. Each bracket will have a production volume of 150,000 per year.

Based on the CES software, we have determined that die casting and hot closed die forging are the two most feasible manufacturing processes for our brackets. These two processes were determined based on the material type (AISI 4620 low alloy steel) and the production volume of 150,000 per year, as well as the shape and thickness of the brackets. Through further research, we have decided to produce these parts through casting because it is cheaper and has a shorter lead-time delivery.

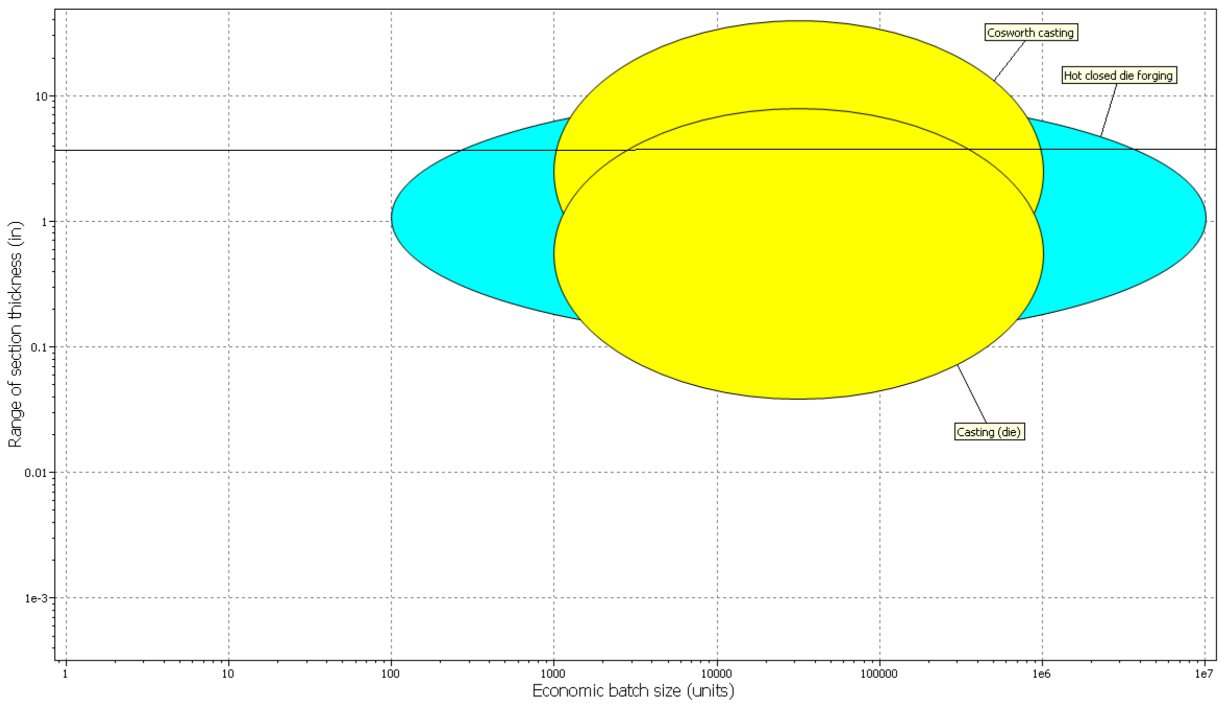

Figure I10. Range of section thickness vs. economic batch size for various manufacturing processes satisfying constraints.